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School of Mechanical Engineering

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Georgia Institute
of Technology
Atlanta, Georgia 30332

GEORGIA TECH 1885-1985

DESIGNING TOMORROW TODAY

THE GEORGE W. WOODRUFF SCHOOL OF
MECHANICAL ENGINEERING

ME 4901
MECHANICAL DESIGN ENGINEERING

NASA/UNIVERSITY
ADVANCED MISSIONS SPACE DESIGN PROGRAM

LUNAR EXCAVATING BUCKET

March 1987

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ABSTRACT

The objective of this project is to optimize the design of a bucket system which will dig soil in the lunar environment. This "bucket system" consists of two inverted clamshell styled buckets which mount to the end effector of "THE LUNAR DIGGER". This bucket design considers the following: soil mechanics, materials, bucket weight, capacity, and geometry. It also looks at optimizing: surface areas, volumes, forces, and weight for the lunar environment.

PROBLEM STATEMENT

To optimize the design of the lunar digger buckets to be as effective as possible in the lunar environment. This redesign will pick up were the initial lunar digger design group left off by placing more effort into the bucket's lunar optimization. This design considers:

MATERIALS:

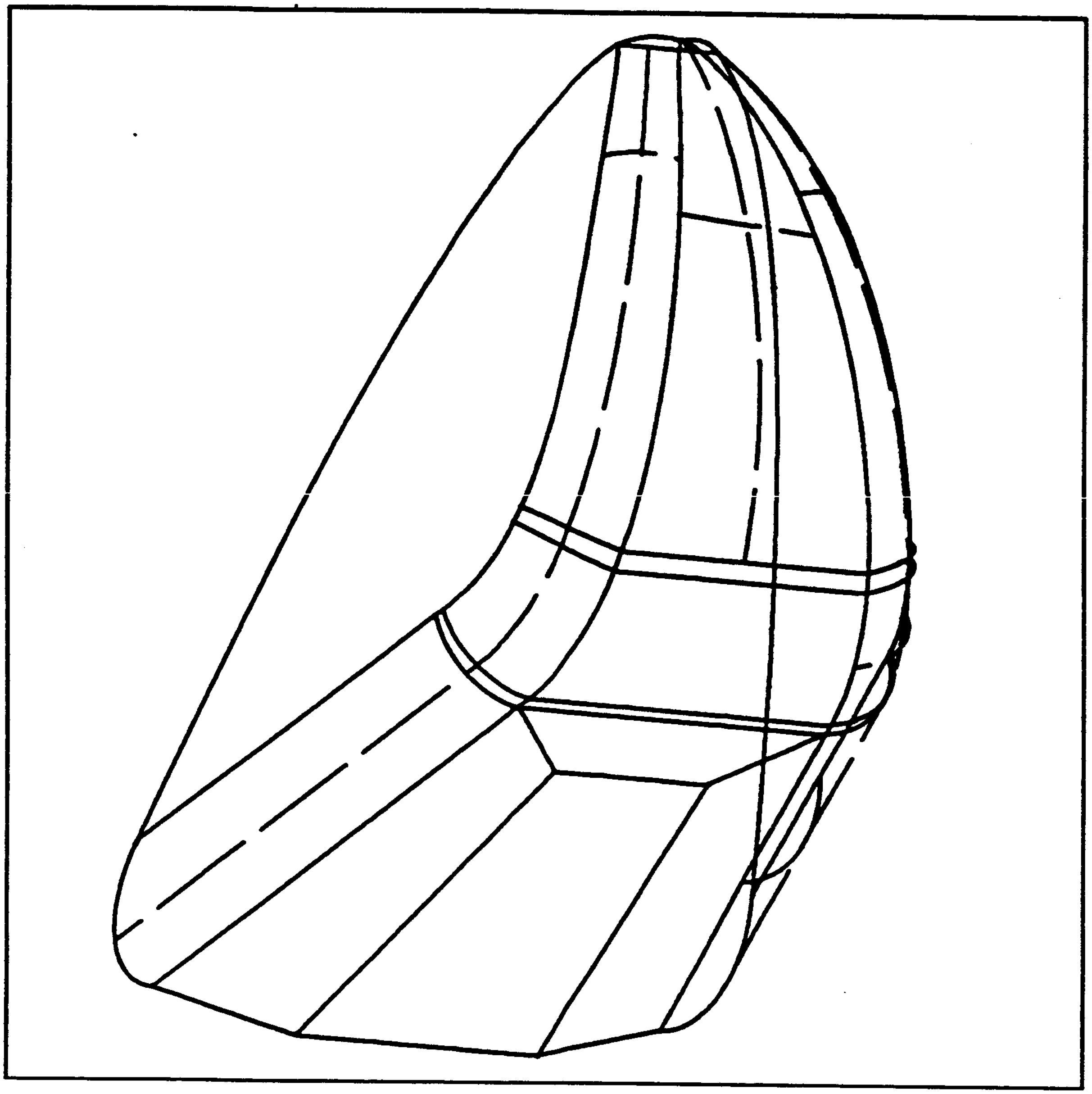
Strength -vs- Weight, Elongation, Wear, Temperature, and Radiation effects.

GEOMETRY:

Volume, Shell Surface Area, Cutting Edge Surface Area, Internal Radius, Capacity, Angle of Cut, and Impact Forces.

SOIL MECHANICS:

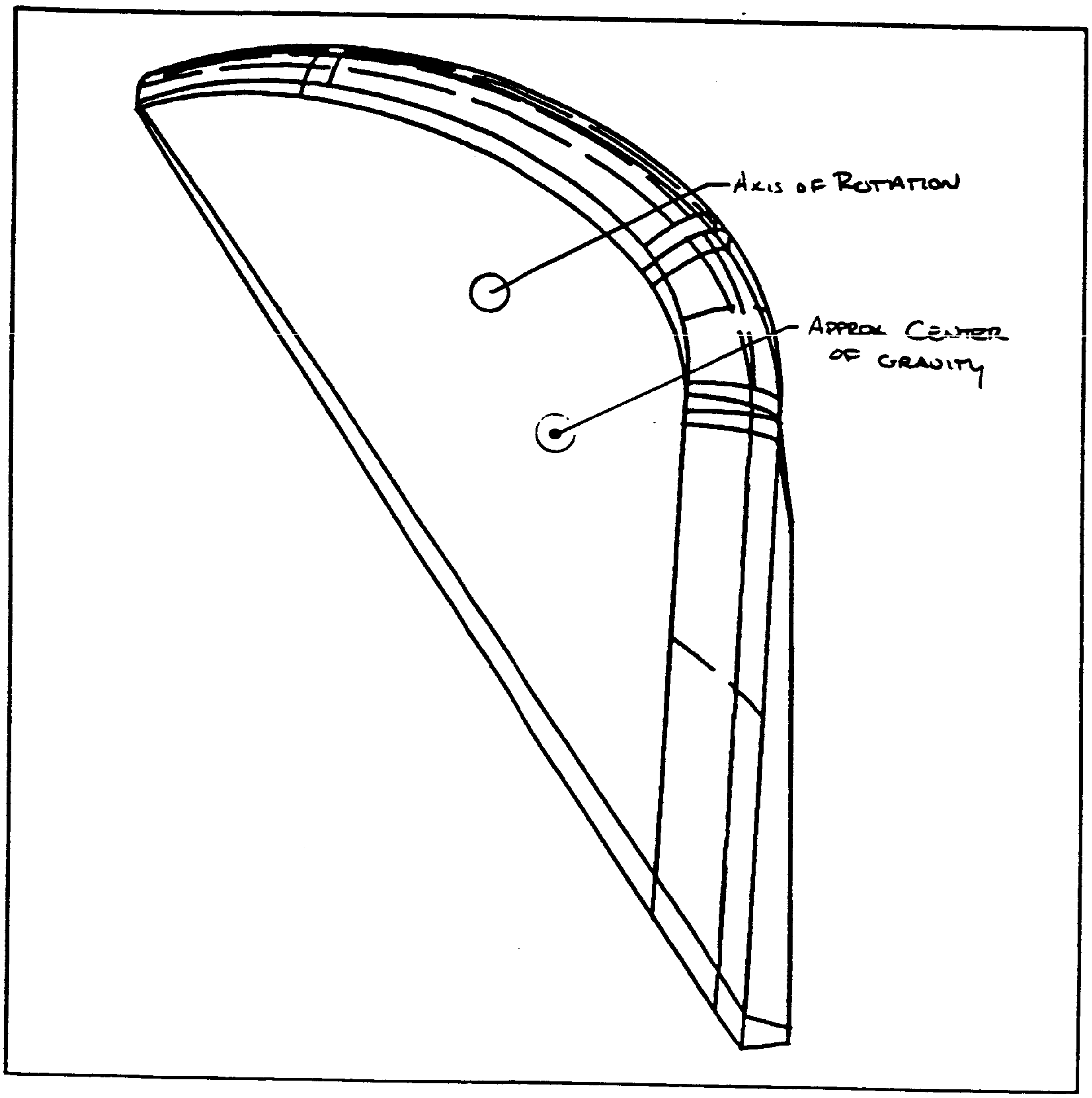
Angle of Cut (internal friction angle), Bearing Capacity, and Rock Size.



CAPACITY .385 FT^3

WEIGHT 59.8 LB

PER BUCKET



AXIS OF ROTATION
APPROX CENTER
OF GRAVITY

DESIGN DETAILS:

GEOMETRY:

The shape of this design is a very critical detail. It involves consideration of volume, surface areas, internal radius, and impact forces.

The benefit of this geometry is that each surface is contoured to flow smoothly to the rear corner area; Thus, allowing digging forces to be distributed toward that point. Essentially, the bucket sides will act as shear planes to distribute the impact and digging forces. An added shear plate will be welded on the rear point, and added support will be placed around the cutting area to reduce digging moments.

This geometry may not fully optimize volume and surface area, but it will tend to fill itself due to its rolling contour, i.e., excess soil will be curled back into the bucket rather than allowed to fall over the sides thus producing a "fuller" bucket.

MATERIAL SELECTION:

The material selected the bucket shell and reinforcement system is aluminum 2014-T6. This material was selected for its strength to weight ratio, elongation characteristics, and weldability. For the bucket teeth and cutting edge, Steel 4140 will be used. This material was selected for its tensile strength, abrasion resistance, impact strength and suitability for high temperature stresses.

FORCE ANALYSIS:

This design had initially intended to analyzed by finite element analysis method, but due to time constraints the force analysis was done by several approximation methods. They are: Johnson column buckling analysis, thin wall pressure vessel approximation, thermal stress analysis, fatigue analysis. Some secondary force calculations were also performed: Shell face friction, impact -vs- digging force comparisons, and full bucket weight calculation. See FORCE ANALYSIS SECTION.

OPTIMIZATION:

This design set out to refine the original design of the digger buckets. In the time frame allowed, full optimization was impossible but the original design was greatly refined. This design increased the capacity of each bucket, while reducing the weight.

PER BUCKET

	Orig. Design	This Design
Capacity:	0.25 yd^3	.385 yd3
Weight:	130 lb	50 lb

Geometry details:

Shell thickness:	.4 cm
Reinforcement thickness	.4 cm
Internal Radius	10 cm
Rear Radius	30 cm

To further optimize this design, I feel that the area which needs the most attention is the dimensions. By developing a relationship between the initial dimensions and the final complex shape, one could vary the dimensions for optimization. A Finite Element force analysis would also help by refining the force analysis which in turn would guide one to other areas of concern.

Bucket Reinforcement: The bucket shell has been designed for ideal force interactions, but actual digging applications will be much different. A bucket reinforcement system has been designed to protect against bucket failure. It will be made of Aluminum 2014-T6 - except for the cutting edge which will be 4140 Steel. This reinforcement outlines the open cutting face to prevent moments and wear, and will be the main support structure around the axis of rotation. It will be welded to the bucket and can be replaced if needed.

Steel Cutting Edge: This will be rivoted to the aluminum bucket shell. Slots which are 2.2 times larger than the holes in the shell will be punched through the steel to prevent warping due to thermal stresses. (The above figure is calculated inthe force analysis section.) The replaceable teeth will be mounted on "fingers" which protrude from this steel cutting edge. See thermal elongation analysis in Force Analysis section and Teeth/ Blade design section.

Teeth Design: To reduce wear on the cutting edge of the bucket and to facilitate soil breaking, replaceable teeth will be mounted to the cutting edge. These teeth will be pinned to the cutting blade which in turn is mounted to the bucket. These teeth will be made of 4140 Steel. See Teeth design section.

Rock Grippers: Since these buckets are required to grasp onto rocks, rock grippers will be needed to make gripping possible. Otherwise, the teflon coating of the buckets will make rock gripping difficult. The grippers , made of aluminum 2014-T6, will be welded to the bottom of the bucket in strategic places. See Rock Gripper section for details.

Axis of Rotation: The ideal axis of rotation for these buckets would be through the bucket's center of gravity. In order to interface these buckets to the digger's end effector, this was not possible - mainly due to actuator positioning problems. The axis of rotation was kept a close as possible to the centroid and still allow the buckets to function reliably. See Reinforcement Section for more details.

CONSTRAINTS

The lunar environment displays some very adverse conditions for construction equipment. Some of these conditions which particularly effect this project are described below.

SOIL MECHANICS

Since this design is to optimize a bucket/soil removal system, the bucket/lunar soil interaction is very critical. Some of the more critical soil characteristics are:

<u>Angle of cut:</u> The cutting of the bucket blade through soil is a very important feature. For lunar soil, the angle of internal friction is 37 degrees. To minimize the forces on the blade tip, an angle of 37 degrees will be designed for the cutting edge. This will not only reduce the normal forces against the blade, but it will also reduce blade wear.

<u>Bearing capacity:</u> This factor of soil mechanics deals with the normal force opposing the bucket blade as the blade shears through the soil. Lunar soil bearing capacity increases with soil depth due to numerous years of compaction.

TEMPERATURES: Due to the diurnal cycles of the moon and it lack of atmosphere, lunar temperatures are very extreme and they range from -300 F to 300 F. Thus, an object sitting partially in the shade will have a temperature of -300 for the shaded portion and a temperature of 300 for the exposed portion -creating very large thermal stresses in the material. See thermal stress analysis in Force Analysis Section for details.

WEIGHT: Due the extreme cost of shipping a material to the moon, weight is a very inportant concern. In order to minimize weight, maximization of other constraints must be considered.

RADIATION: The main consideration about radiation is its affect on materials. Reseach showed that radiation effects on metals, in general, are very small - almost no effect on physical properties at all.

DISCUSSION OF DESIGN

The design process for these buckets took a very structured approach. First, the design was broken down into functional requirements and parameters. These can be easily viewed on the last page of this report. The stated functional requirements are located below as sub-headings, which in turn are reduced to parameters.

MATERIALS:

Strength -vs- Weight Ratio: Very important criteria of a structural material, this ratio indicates the materials which are extremely strong for their densities.

Temperature: As stated above, the lunar environmemt displays very large temperatures variations and anything working in this environment must withstand these fluctuations. Finding information on positive temperatures was very easy, but data on negative temperatures was almost non-existant. Thus, finding a suitable material became quite a challenge.

Material Wear: Lunar soil is essentially very gritty and coares. It produces extreme wear on any exposed surface and can easily cause failure of poorly designed equipment. These buckets will be coated with Teflon for overall wear resistance, and the designed replaceable teeth will absorb wear on the cutting surface.

Elongation: This material characteristic in very important for structural lunar materials. High temperature ranges and forces, can easily cause a part to fail if the materials elongation is not considered. See thermal stress calculation in Force Analysis section.

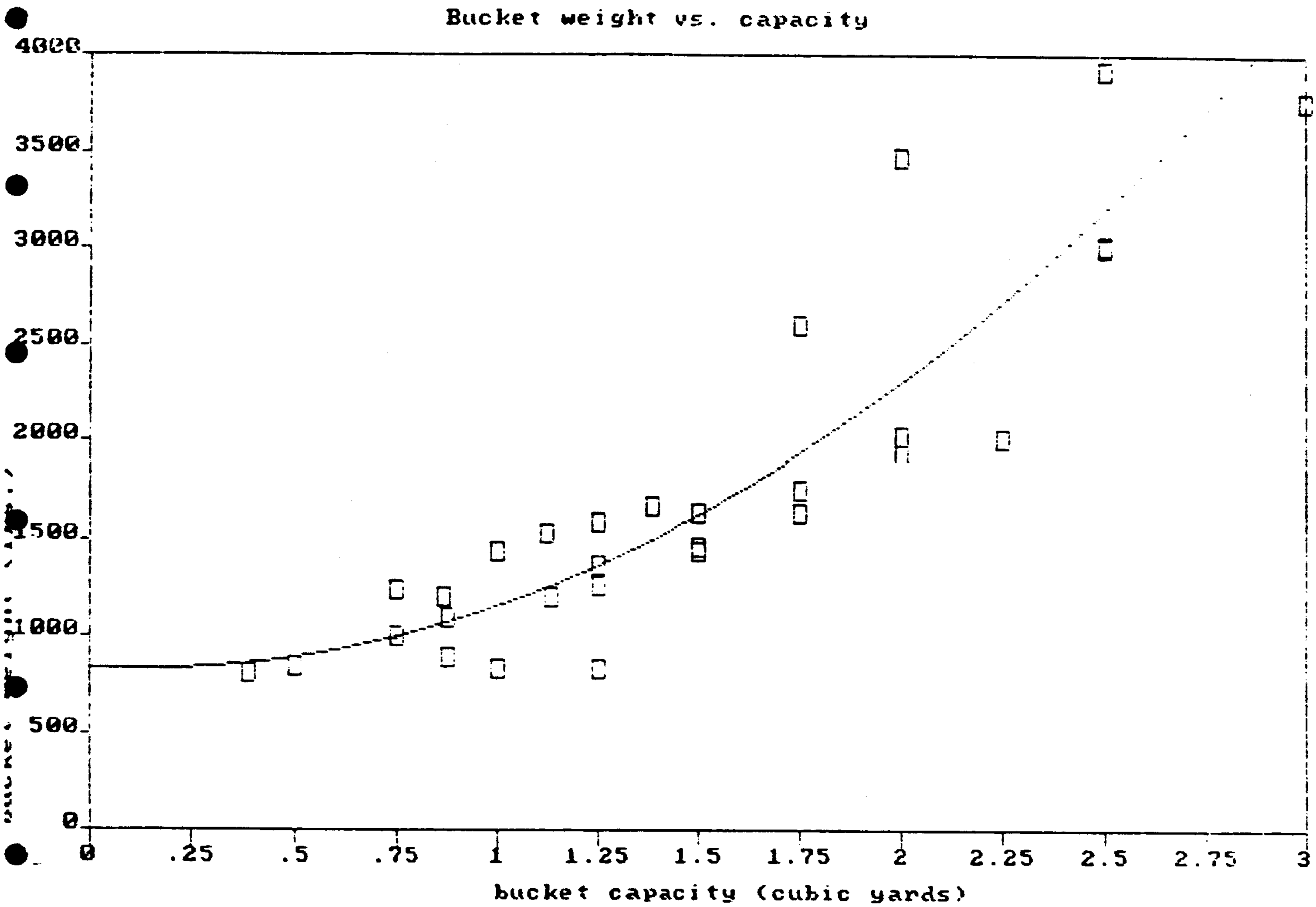
GEOMETRY:

Bucket geometry consists of the overall shape of the buckets. This shape can be broken down into optimum dimensions, internal radii, capacity, and weight.The basic geometry that was used for this project was found in a book of roman agricultural equipment - but it was altered for this application. Research was done in this area, for ancient agriculture tools were refined by trial and error and the user was very concerned of its effeciency.

Dimensions: From research of different bucket shapes and physical modeling of feasible shapes, the final bucket dimensions were defined. There is no hard and fast reasoning for these dimensions, but from practical analysis and definition of parameters the dimensions were decided. The enclosed drawings used a standard scale so that scaling of the final shape would be simple.(See page 15)

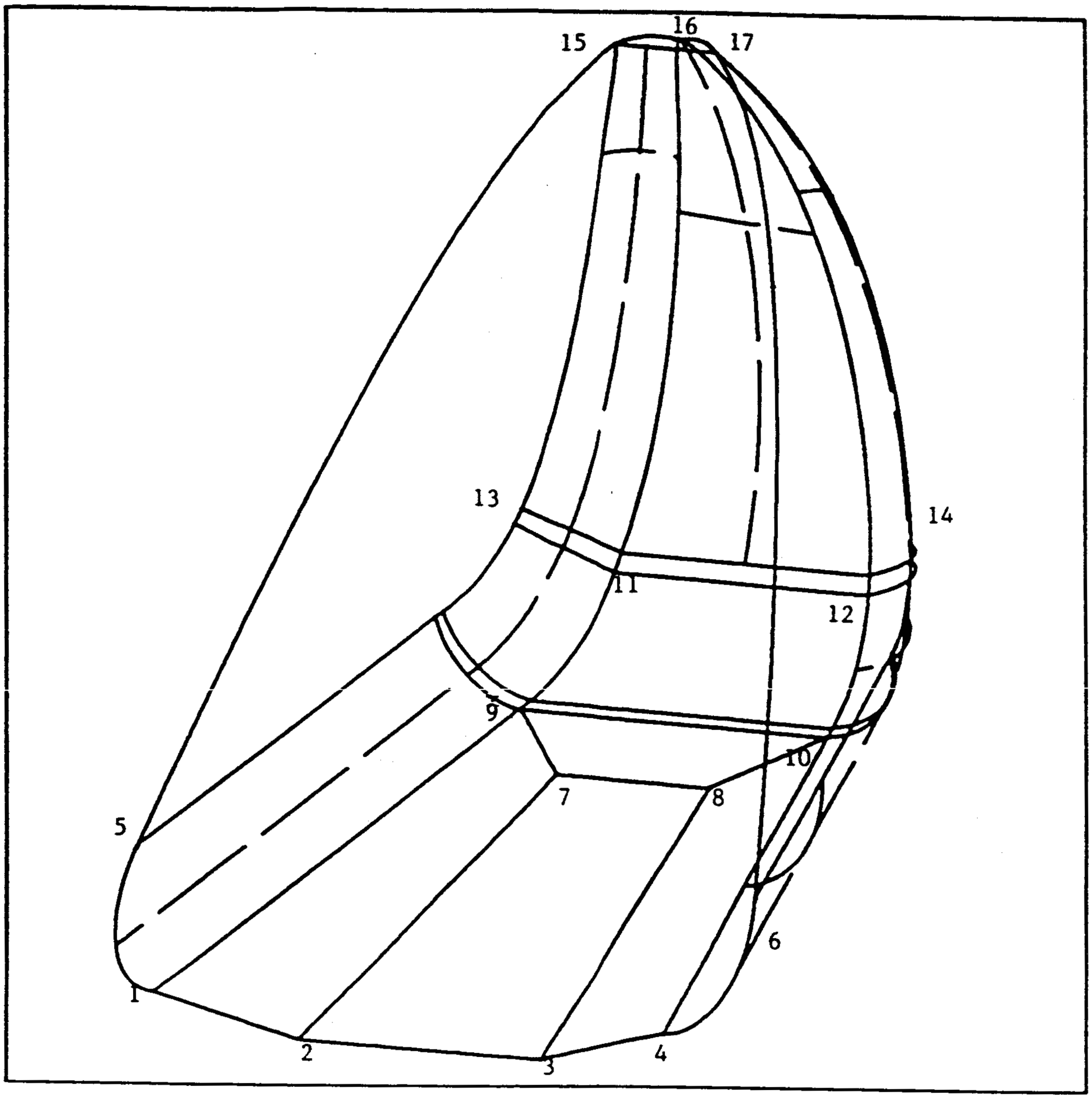
Capacity: From a graph of Weight -vs- Capacity, the lunar digger design group from Fall 1986 found that a total capacity of 0.5 cubic feet was optimum and feasible for this design (see page 14). Thus, the dimensions of the enclosed drawing can be scaled to fit this capacity.

geometry



THE REGRESSION POLYNOMIAL OF LINE 1 -

(8.401E+02) + (-3.051E+02)*X + (3.752E+03)*X´ 2
THE VARIANCE - 1.477E+05



Bucket Points (relative to non shown origin)

	X	Y	Z
1)	75.	15.6	0.0
2)	75.	33.66	-5.0
3)	75.	66.34	-5.0
4)	75.	84.40	0.0
5)	60.65	8.35	10.66
6)	60.65	91.65	10.66
7)	18.6	39.	-2.0
8)	18.6	60.	-2.0
9)	8.6	29.2	0.0

	X	Y	Z
10)	8.6	70.8	0.0
11)	-11.91	33.04	6.69
12)	-11.91	66.96	6.69
13)	-9.44	23.73	15.92
14)	-9.44	76.27	15.92
15)	-22.73	44.225	72.64
16)	-22.73	56.775	72.64
17)	-26.27	49.62	71.64

CENTER OF GRAVITY:

X= 15.6, Y= 50, Z= 22.46

Weight: To minimize bucket weight, bucket surface area must be minimzed. Weight of these buckets are extremely important for shipping reasons and mostly to minimize the excess weight needed for the articulated digger arm to be carrying.

Manufacture: Since the shape of these buckets are very complex, manufacture will be difficult. The specified material is very weldable, which makes manufacture possible and allows for maintenance on the lunar surface.

Internal Radius: Lunar soil is essentially very sticky and it fill any voids on the bucket surface - requiring the digger to carry around extra weight. It is for this reason that internal radii are used, for they reduce the number of places that soil can accumulate.

* The optimum shape for the bucket is one that will loosen any accumulated soil from the bucket each time a dig is made.

FORCE ANALYSIS:

This section explains the reasoning behind the important force calculations - why certain forces were used for specific uses.

Column Buckling: Since the maximum force exerted on the bucket will probably be during initial impact with the soil, this force was analyzed first. By assuming impact forces would tend to buckle the bucket, column buckling approximations were performed. Initially, an average channel cross-section was located to simulate the bucket cross-section. Then, the J.B. Johnson equation was applied and a thickness for the bucket material was found. With this thickness, the moment inertia for the actual bucket cross-section was found and reapplied to check for accuracy. This force value from the reapplication checked within 1.5%, so this approximation should be accurate.

Thin Wall Pressure Vessel Approximation: Assuming the the actual digging forces will act on the bucket face with a constant pressure, the bucket can be analyzed for shear stress in the bucket shell by this method.

Thermal Stresses: Due to the temperature range on the moon, thermal stresses in materials are very critical. These, along with bucket shell stresses can easily cause failure of the part. Thermal stresses are also very important for the connection of the steel cutting blade to the shell. Thermal stresses vary relative to material elongation which is very important for this connection.

Fatigue: Since these buckets will be required to withstand a great deal of use, a fatigue analysis is necessary. The results can be found in the Force Analysis section and show that due to the relatively small forces, fatigue is not really a factor.

Conclusion:

From this project, I feel that I greatly increased my own design skills, and the design of the Lunar Excavating Bucket. The results that I obtained for refinement, a 54% increase in capacity with a 54% decrease in weight, seemed reasonable for a school quarter's of work. I know that there are still some vague areas of this report, but with the time alloted I covered as many details a possible.

Parts List/ Bucket:

Quantity	Description	Weight	Cost
1	0.4 cm Thick Bucket Shell	40.0	8000
7	Bucket Teeth	3.0	2000
1	Reinforcement System	5.0	3000
1	Teflon Coating	0.1	1000
3	Cutting Blade Section	10.3	5200
10	Pins for Teeth	0.2	100
17	Rivots of Blade	0.4	150
3	Rock Grippers	0.8	1000
	Totals	59.8 lb	$ 20450

BIBLIOGRAPHY:

Askeland, Donald; The Science & Engineering of Materials; Monterey, Ca.; Brooks 1984

Bachus, Robert; personal interivew on 1-29-87 at Georgia Institute of Technology.

Baker, Sir John; The Design of Design; University Printing House, Cambridge, 1969

Deiter, George; Engineering Design; McGraw Hill, N.Y. 1983

Fox, Richard; Optimization methods of Engineering; Addison-Wesley Publishing; London, 1971

Heller, Martin E.; Metal Selector program; American Society of Metals; 1985

Kalpakjian, Serope; Manufacturing Processes for Engineering Materials; Addison- Wesley Publishing, London 1984

Lunar Digging Appartus, The Final Report; Fall 1986; Georgia Institute of Technology; ME 4182

Meyers, Carolyn; Personal interview on 1-20-87 at Georgia Institute of Technology

NASA Document, Appolo 14, Preliminary Science Report; 1971

NASA Document, NAS8-2631, Determination of Low Temperature Fatigue Properties of Structural Metal Alloys, 10/64

Pagonis, George; The Light Metals Handbook; Van Nostrand, Toronto 1054, Vols. 1 & 2

Prodolskiy -A-D; Lunar Exploration- Lunik 16 Lunar Probe, microfiche at Price Gilbert Library, N72-24927-7200

Rittenhouse, John & Singleton, J.B. ; Space Materials Handbook; NASA 3rd., 1963

Shigley, Joseph & Mitchel, Larry; Mechanical Engineering Design; McGraw Hill, N.Y. 1983

Sowers, G.; Intro. to Soil Mechanics and Foundations; MacMillan Publishers, N.Y. 1979

Thompson, G.; Materials in Space Technology; ILIFFE Books Ltd., British Interplanetary Society, 1963

White, K.D.; Agricultural Implements of the Roman World; Cambridge Press,1967

APPENDIX:

MATERIAIS

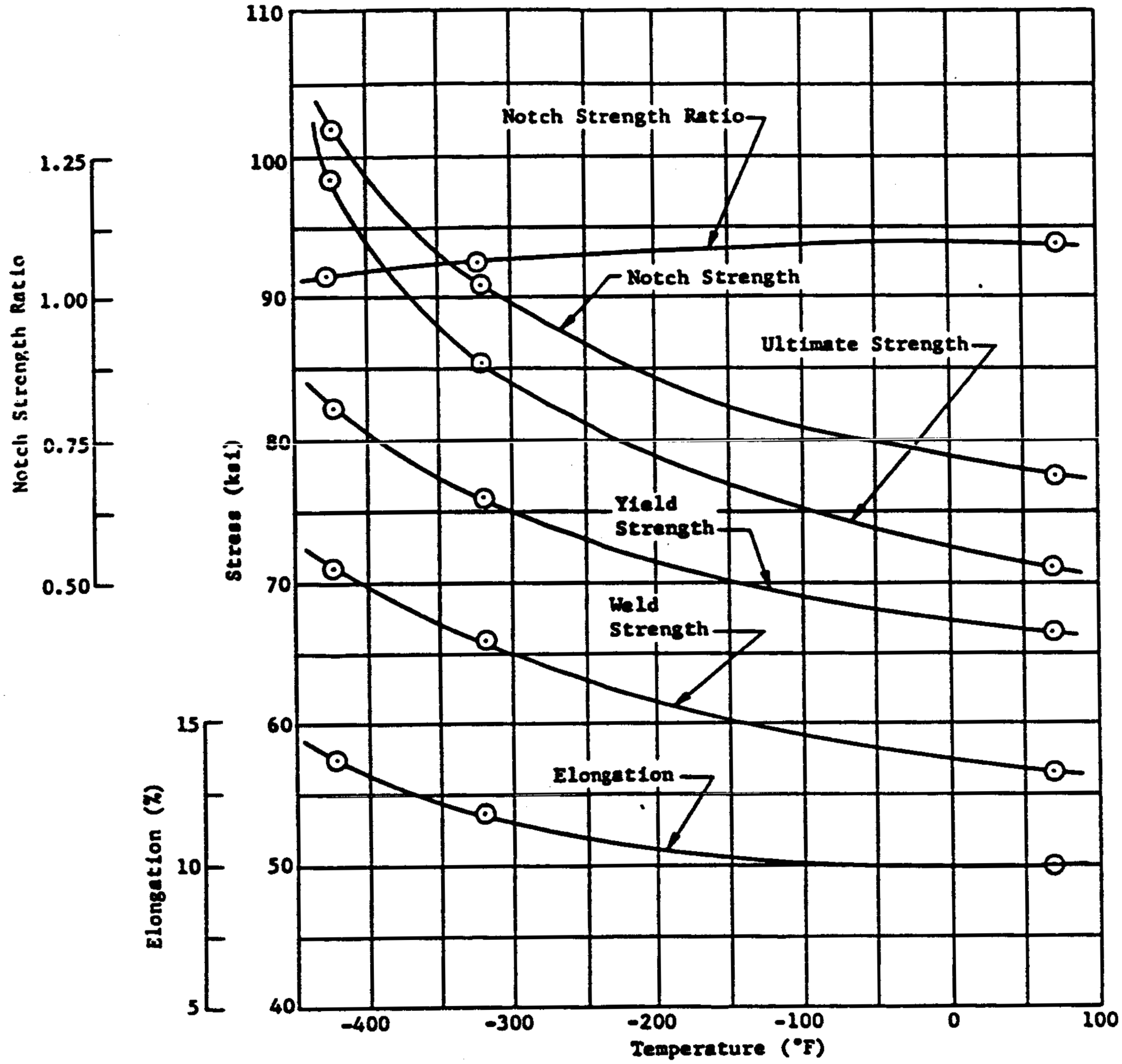


Fig. 4 Tensile Properties of 2014-T6 Aluminum Alloy at Cryogenic Temperatures

Table B-1 Tensile Properties of 2014-T6 Aluminum Alloy at Cryogenic Temperatures

Temperature (°F)	Ultimate Strength (ksi)	Yield Strength, 0.2% Offset (ksi)	Elongation, in 2-in. (%)	Modulus of Elasticity, (psi x 10^6)	Weld Strength (ksi)	Notch Strength* (ksi)	Notch Strength Ratio
70	71.8 70.4 70.9 71.0 avg	67.3 66.3 65.9 66.5 avg	10.5 10.0 9.5 10.0 avg	10.6	55.1 55.3 59.0 56.5 avg	77.0 77.7 77.8 77.5 avg	1.09
-320	85.3 84.9 85.9 85.4 avg	76.4 75.8 75.8 76.0 avg	11.5 12.0 12.0 11.8 avg	11.7	66.2 64.6 67.0 65.9 avg	91.2 91.2 90.4 90.9 avg	1.06
-423	100.0 99.0 96.3 98.4 avg	83.0 82.8 80.7 82.2 avg	14.0 13.5 13.5 13.7 avg	12.3	74.3 72.2 69.5 68.5 71.2 70.5 71.1 avg	102.0 102.5 101.0 101.9 avg	1.04

*K_t = 3.5.

25

a. 2014-T6

The unnotched 2014-T6 alloy exhibits increased strength for 10^6 cycle life of approximately 300% with reduction in temperature from 70 to -423°F. As noted for static behavior, strengthening is greatest between -320 and -423°F. The fatigue ratio* increased from 0.23 to 0.48 with reduction in temperature. The curves obtained at 70 and -423°F were significantly flatter than the -320°F curve which showed a slight knee. Although the reason for this is not completely known, a slight misalignment may be responsible for this condition. Testing performed on similar aluminum alloys at -320°F during the second year's effort did not show this behavior. Alignment accuracy achieved during the latter period is believed to be superior to that attained during the first year.

Notch test results show a significant decrease in fatigue strength compared to the unnotched data. The fatigue strength reduction factor (K_f)†, also known as fatigue notch factor, was rather poor. At 10^6 cycles, the factor increased from 2.0 at 70°F to 6.0 at -423°F. The low cycle (10^3) end of the curve showed a less significant effect with an initial value of 1.4 at 70°F, which increased to almost 2.0 at low temperatures.

Welded joints also show a loss of strength compared to the unnotched material. However, comparing the 70°F static 2014-T6 weld joint efficiency (80%) with the ratio of weld fatigue strength/unwelded fatigue strength (69%), a very good retention of weld strength under dynamic loading is evident. At cryogenic temperatures, a marked decrease in this strength ratio is noted.

---

*Fatigue ratio = $\dfrac{S_u \text{ (static tensile strength)}}{S_n \text{ (fatigue strength at n cycles)}}$.

†Fatigue strength reduction factor or fatigue notch factor -

$$(K_f) = \frac{\text{Fatigue strength of unnotched specimens at n cycles}}{\text{Fatigue strength of notched specimens at n cycles}}.$$

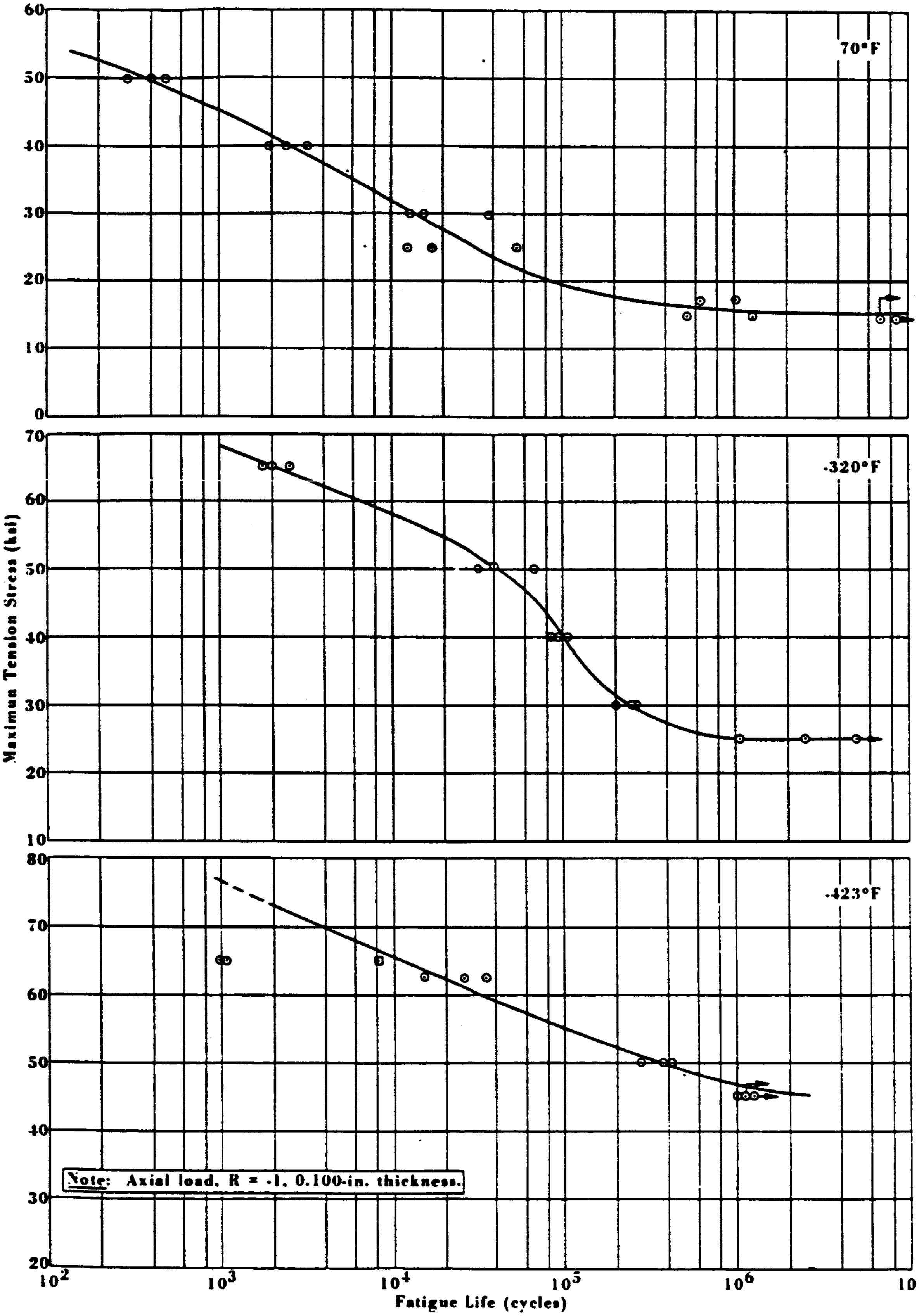


Fig. 28 Fatigue Properties of Unnotched 2014-T6 Aluminum Alloy

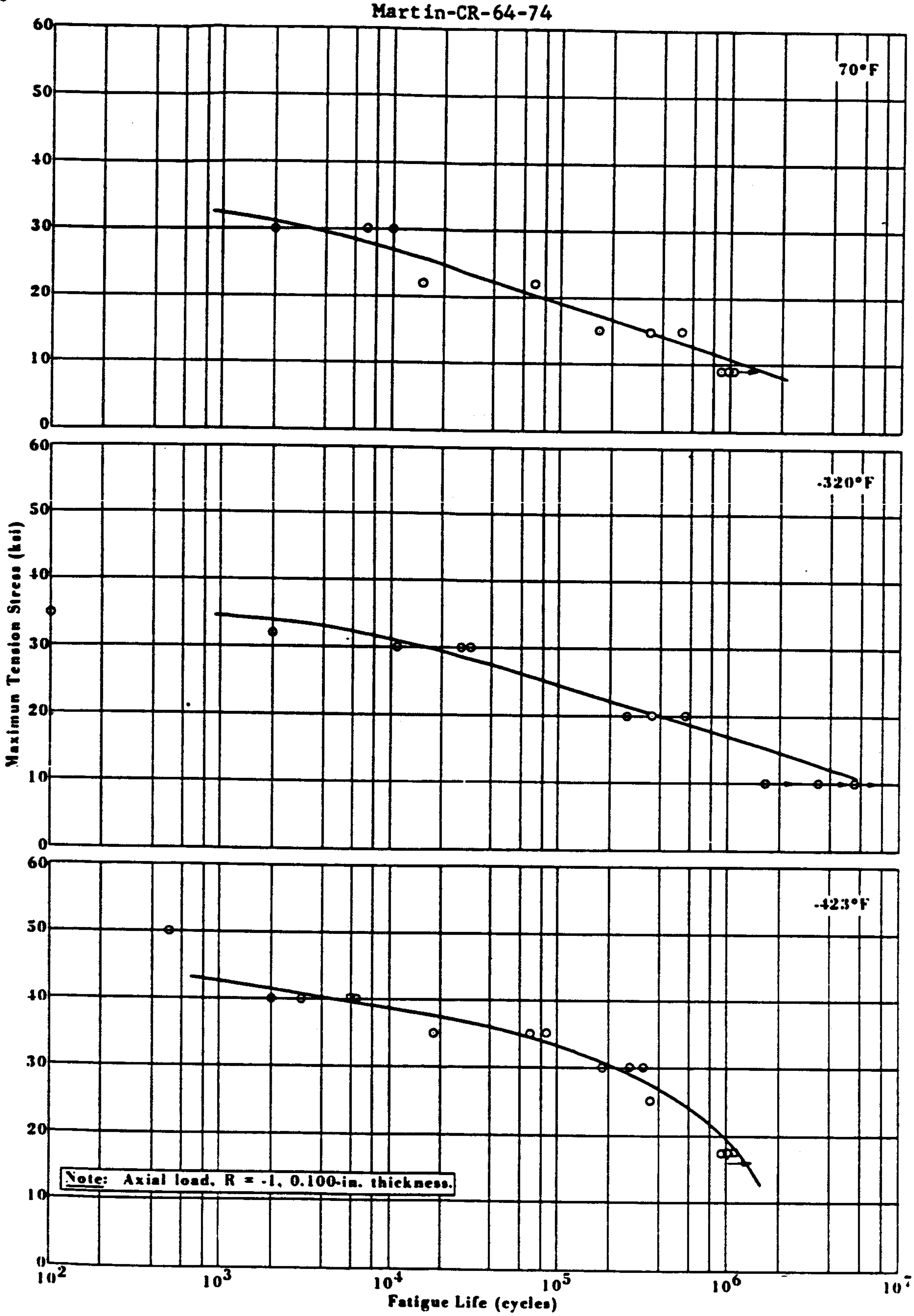


Fig. 30 Fatigue Properties of Welded 2014-T6 Aluminum Alloy

Table C-3 Fatigue Properties[a] of Welded 2014-T6 Aluminum Alloy

Temperature					
70°F		-320°F		-423°F	
Maximum Tension Stress (1000 psi)	Cycles to Failure	Maximum Tension Stress (1000 psi)	Cycles to Failure	Maximum Tension Stress (1000 psi)	Cycles to Failure
30.0	2.00×10^3	35.0	1.00×10^2	50.0	5.00×10^2
30.0	7.00×10^3	32.0	2.00×10^3	40.0	2.00×10^3
30.0	1.00×10^4	30.0	1.05×10^4	40.0	3.00×10^3
22.0	1.50×10^4	30.0	2.74×10^4	40.0	6.00×10^3
22.0	2.10×10^4	30.0	2.75×10^4	40.0	6.00×10^3 [b]
22.0	6.90×10^4	20.0	2.51×10^5	35.0	1.80×10^4
15.0	1.70×10^5	20.0	3.42×10^5	35.0	6.80×10^4
15.0	3.29×10^5	20.0	5.51×10^5	35.0	8.40×10^4
15.0	5.14×10^5	10.0	1.59×10^6 (disc)	30.0	1.82×10^5
9.0	1.02×10^6 (disc)	10.0	3.30×10^6 (disc)	30.0	2.65×10^5
9.0	1.03×10^6 (disc)	10.0	5.29×10^6 (disc)	30.0	3.18×10^5
9.0	1.03×10^6 (disc)			25.0	3.44×10^5
				17.5	1.01×10^6 (disc)
				17.5	1.03×10^6
				17.5	1.06×10^6

a. Axial load R = -1.
b. Specimen previously run at 17,500 psi for 1.01×10^6 cycles without failure.

A l l o y   D a t a

DSG G41400 A Common 4140 User JIS G4052 SCM4H
BS 970 708A37 DIN 1.7223 AFNOR 35CD4 Class ALLOY STEEL
Temper/condition ANNEALED

Tensile strength	95	KSI	Hardness	197	HB
Yield strength	61	KSI	User rating	0	
Elongation	25.5	%	Reference	4 . 8	

Formability E Weldability E Machina'ty C Hardena'ty C Availa'ty B Proc. cost
Available as WROUGHT FORGING SHEET STRIP PLATE BAR WIRE
Props
Comments: A chromium-molybdenum, medium-carbon steel with high hardenability and good fatigue, abrasion, and impact resistance. Used where service conditions are not severe enough to require 4340 steel. Tensile strengths up to 240 ksi are readily achieved through conventional heat treatment. Nitriding for maximum wear and abrasion resistance, 4140 is a deep-hardened alloy suitable for high-temperature stresses, or combinations of such stresses in small and large sections. When fully hardened, demonstrates the outstanding property of relatively high impact strength at high-hardness, tensile strength. Uses include small gears, pinions, and ball studs, and for high-strength bolts, cap

F1=HELP F2=PRINT F3=FILE ↵=SELECT Esc=QUIT Home PgUp PgDn (comments)

A l l o y   D a t a 1

DSG G41400 A Common 4140 User JIS G4052 SCM4H
BS 970 708A37 DIN 1.7223 AFNOR 35CD4 Class ALLOY STEEL
Temper/condition ANNEALED

Tensile strength	95	KSI	Hardness	197	HB
Yield strength	61	KSI	User rating	0	
Elongation	25.5	%	Reference	4 . 8	

Formability E Weldability E Machina'ty C Hardena'ty C Availa'ty B Proc. cost
Available as WROUGHT FORGING SHEET STRIP PLATE BAR WIRE
Props
Comments: screws, and socket- and recessed-head screws. These products can be heat treated after machining or forming.

F1=HELP F2=PRINT F3=FILE ↵=SELECT Esc=QUIT Home PgUp PgDn (comments)

FORCE ANALYSIS

MAX FORCE:

TRIPOD	2000 lb
Digger Arm	1000 lb
	3000 lb EARTH
	≈ 500 lb MOON

$$\text{Mass}_{moon} = 500/5.36 = 93.16 \text{ slugs}$$

$$F_{max} = Ma.$$

LET $a = 30 \text{ ft/sec}^2$

[IMPACT Force required to accellerate vehicle 30 ft/sec² into atmosphere]

$$F = (93.16)(30 \text{ ft/s}^2) \cong 2800 \text{ lb}_M$$

Column Buckling Approx:

CONVERSION FACTOR
TO fit TABLE A-11 (Shigley, Machine Design p803
a = 9
b = 1.72

$$\frac{90}{4} = \frac{x}{Y}$$

$$\frac{P_{cr}}{A} = S_y - \left(\frac{S_y}{2\pi}\right)^2 \left(\frac{l}{k}\right)^2 \frac{4}{E}$$

$$\frac{P_{cr}}{A} = 32k - \left(\frac{32K}{2\pi}\right)^2 \left(\frac{4.44}{.45}\right)\left(\frac{4}{10.3E6}\right)$$

$E = 10.3 \times 10^6$ psi $= 7.1E10$ N/m²
$S_y = 32$ kpsi $= 2.41E8$ N/m²

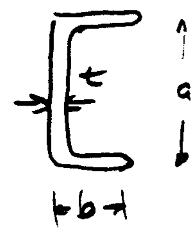
$$\frac{P_{cr}}{A} = 32000 - 99.4$$

IF $Y = 100$, $x = 4.4 = l$

$$\frac{P_{cr}}{A} \cong 31900$$

$$\frac{P_{cr}}{A} = 31{,}900 \quad , P_{cr} = 2800$$

$$\Rightarrow A = \frac{2800}{31900} \approx .0877$$



$$1.72t' - t'^2 + 4t' = A$$

$$t'^2 - 5.72t' + .0877 = 0$$

$$\Rightarrow t' \approx .016$$

$$\frac{90}{4} = \frac{t}{t'} = \frac{t}{.016} = .36 \text{ cm}$$

USING (t) of MATH PACK OF HP. -41CX TO CALCULATE New Moment Inertia

$$P_{cr} = \frac{C\pi^2 EI}{l^2}$$

$$C = 1/4$$

- Area = 568.1 cm²

$I_x = 72.67\text{E}3 \text{ cm}^4$
$I_y = 674.5\text{E}3 \text{ cm}^4$

$I_x = 7.2 \times 10^{-8} \text{ m}^4$

$$P_{cr} = \frac{.25\pi^2(7.1\text{E}10)(7.2\text{E}-8)}{1^2}$$

$$= 12613.4 \text{ N} \approx \underline{\underline{2834.46 \text{ lb}}}$$

$P_{cr_{INPUT}} = 2800 \text{lb}$

$P_{cr_{OUT}} = 2834.46$

CHECKS calculated "t" and CRITICAL FORCE

NATIONAL 42-381 50 SHEETS 5 SQUARE 42-382 100 SHEETS 5 SQUARE 42-389 200 SHEETS 5 SQUARE

NATIONAL 42-381 50 SHEETS 5 SQUARE 42-382 100 SHEETS 5 SQUARE 42-389 200 SHEETS 5 SQUARE

Weights

SHELL AREA $= 16.33 \times 10^3\ cm^2$

$$\text{SHELL Volume} = \overset{t\,(cm)}{t}\left(16.33 \times 10^3\ cm^2\right)\left(\frac{1m}{100cm}\right)^3$$

Weight of Aluminum $= 26.6\ KN/m^3$

$$\text{SHELL WEIGHT} = \overset{\text{in cm}}{t}\left[4344\ N\right]$$

$1\,KN = 1000\,N$ $\quad$ $4.45\,N \sim 1\,lb$

$$\text{SHELL WEIGHT} \sim t\left[97.51\ lb\right] \quad (lbs)$$

$t = .36\ cm$, use $.4$ $\Rightarrow$ SHELL WEIGHT $= \underline{\underline{39\ lb}}$

Volume Conversions

$$cm^3 = \left[1.3079 \times 10^{-6}\right]\ yd^3$$

FROM CATIA

Volume $= .209E6\ mm^3 = .273\ yd^3$

Cone Volume above shell $\approx$ Open Shell face Area.

Approx. with simple cone volume

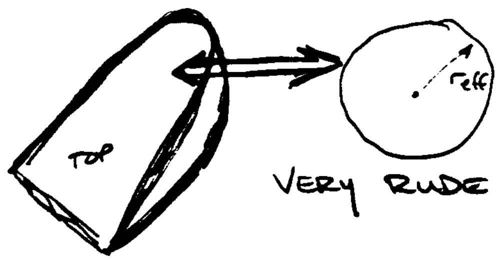
Volume Calculation

Calculated Volume + Cone Volume

Approx

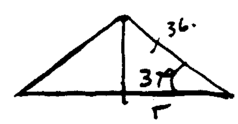
Top Bucket Surface Area = Base cone surf. Area

$S.A = \pi r_{eff}^2$



Very Rude Approx.

Angle of Internal friction = 37°



$r = x \cos 37°$

$x \sin 37° = h$

Cone Vol $= \dfrac{\pi r^2 h}{3}$

From CATIA, Open face surface Area $= 7.47 \times 10^3$ cm²

$S.A = 7.47 \times 10^3$ cm²

$\Rightarrow r_{eff} = 48.76$ cm

$h = 36.74$

Cone Vol = 91495. cm³ = .1196 yd³

Total Vol = 2.95×10^5 cm³ =

$\Rightarrow$ **.385 yd³**

Full Bucket Weight = $.295 \times 10^5$ cm³ (2 gm/cm³)

= 59 kg (1.635 m/s²) = 96.465 N

= 21.67 lb of Soil + Bucket weight ≈ **70 lb**

Thin Wall Pressure Vessel Approx

[From Shigley, Machine Design]

$$\sigma_t = \frac{pd}{2t}$$

$p \equiv$ pressure

d_i = inside diameter x 2m

t = thickness

p/Depth = 6575 Nm/m² [From Soil/Bucket Force Analysis]

$$p = 6575 \text{ N/m}^2 \underbrace{(.9)}_{\text{Depth}} = 5920 \text{ N/m}^2$$

$$\sigma_t = \frac{(5920 \text{ N/m}^2)(4 \text{ m})}{(.0036 \text{ m})} = 6.57 \times 10^6 \text{ N/m}^2$$

Thermal Stresses

Coeff. of Thermal Expansions	
Aluminum	13.3×10^{-6} 1/°F
Steel	6×10^{-6} 1/°F

$$\sigma = \alpha(\Delta T)E$$

$E_A = 7.1 \times 10^{10}$ N/m²

$E_S = 207 \times 10^9$ N/m²

$\Delta T = 600$ °F

$$\sigma_A = (13.3 \times 10^{-6})(600)(7.1 \times 10^{10})$$

$$= \underline{5.666 \times 10^8 \text{ N/m}^2}$$

$$\sigma_S = (6.0 \times 10^{-6})(600)(207 \times 10^9) = \underline{7.452 \times 10^8} \text{ N/m}^2$$

WALL STRESS + THERMAL STRESSES

$$\sigma_t + \sigma_A = \sigma_{TOTAL}$$

$$6.57 \times 10^6 + 5.666 \times 10^8 = \underline{5.7317 \times 10^8 \ N/m^2}$$

$$\sigma_{TOT} \le Ee = 7.1 \times 10^{10} (.08) = \underline{5.68 \times 10^9} \ N/m^2 \ \checkmark$$

Although these figures are very close, these buckets will have a teflon coating and accumulated soil, which will tend to reduce thermal stresses. Also, conduction through the material will reduce thermal gradients.

Fatigue Analysis

Aluminum

Max Stress on Bucket

2800 lb ≈ 12500 Nt

Smallest bucket area = (.8m)(.0036m) = .00288 –

$$\sigma = \frac{12500}{.00288\ m^2} = 4.34\times10^6\ N/m^2 \left(\frac{1\ psi}{6890\ Pa}\right) \Rightarrow 629.9\ psi$$

From Fatigue Data in Materials Section of Appendix, Fatigue is not a problem.

Elongation due to thermal stresses.

$\sigma_{alum} = 5.66\times10^8\ N/m^2$ $\qquad \sigma = Ee \qquad e = \frac{\sigma}{E}$

$\sigma_{STEEL} = 7.452\times10^8\ N/m^2$

$E_{AL} = 10.3\times10^6$

$E_{ST} = 30\times10^6$

$$e_A = \frac{5.66\times10^8}{10.3\times10^6} = 54.95$$

$$e_{ST} = \frac{7.452\times10^8}{30\times10^6} = 24.84$$

$$\frac{e_{ST}}{e_{AL}} = \frac{24.84}{54.95} = .452 \qquad \frac{1}{.452} = 2.2$$

⇒ For each cm that the aluminum elongates, the steel will elongate .452 cm. This factor will be incorporated into the Steel Blade / Alumi bucket interface.

Bucket Soil Interaction

Submerging the Bucket:

Q_{EB} = End bearing shear

Q_{sf} = skin friction shear

$$Q = Q_{EB} + Q_{SF}$$

B = width
C = cohesion
z = submerged depth
γ = density

$$q_{eb} \cdot (A) = Q_{EB}$$

$$q_{eb} = \frac{B \gamma N_\gamma}{2} + C N_c + q_q N_q$$

For small B, $q_{eb} = C N_c + q_q N_q$

From Fig 11.11 (Sowers)

$N_c = 60$

From lunar Soil data $N_q = 35$

$$C = \text{cohesion} = (.5\ \text{psi})(6890\ \text{Pa/psi}) = 3445\ \text{N/m}^2$$

$$q_q = \gamma z = (2000\ \text{kg/m}^3)(1\ \text{m}) = 2000\ \text{kg/m}^2$$

$$(2000\ \text{kg/m}^2)(1.365\ \text{m/s}^2) = 2402.4\ \text{N/m}^2$$

$$q_{eb} = 3445(60) + 2402(35) = \underline{290\ \text{KN/m}^2}$$

$$\text{cross-sectional Area} = 568.1\ \text{cm}^2 = .05681\ \text{m}^2$$

$$\Rightarrow Q_{EB} = 16.52\ \text{KN} \approx \underline{3720\ \text{lb}}$$

42-381 50 SHEETS 5 SQUARE
42-382 100 SHEETS 5 SQUARE
42-389 200 SHEETS 5 SQUARE
NATIONAL

Skin Friction

$$C_a = .9C = .9(3445) = 3100$$

$$Q_{sf} = (S.A.)\, q_f \qquad q_f = C_a + \sigma_n' \tan\delta$$

$$\sigma_n' = K_s(\gamma_s z_s + \cancel{\gamma_w z_w}) \quad 0 \text{ = No water on moon}$$

$$K_s = 3 \quad \text{table 11.2 (Sowers)}$$

$$\sigma_n' = 3(2000\ kg/m^3)(1.635\ m/s^2)(1m) = 9810\ N/m^2$$

$$\phi = 37^\circ = \delta$$

table 11.3 Sowers

$$q_f = (3100\ N/m^2) + (9810\ N/m^2)(.2)$$

$$= \underline{5062\ N/m^2}$$

Max Surface Area $= 22.94 \times 10^3\ cm^2 = 2.294\ m^2$

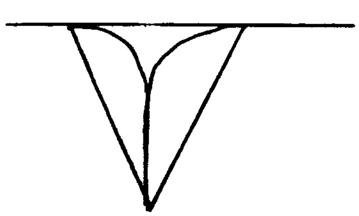
$$Q_{sf} = (5062)(2.294) = 11612\ N$$

$$Q_T = Q_{SF} + Q_{EB}$$

$$16.52\ kN + 11.61\ kN = \underline{28.13\ kN}$$

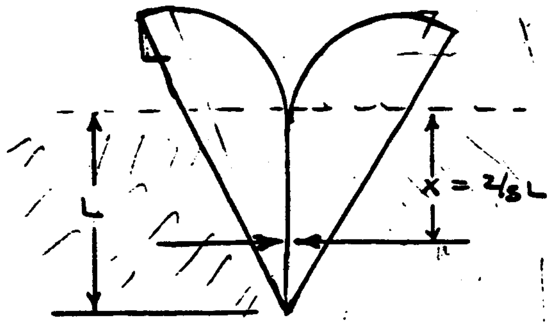
$$\approx \underline{6321\ lb}$$

This Force Analysis calculates the force required to fully submerge the buckets into the lunar soil. But, the bucket will not be used in this manner - they will scoop "surface soil" rather than fully submerged soil.



Fully submerged buckets

Actual digging practice.



$\gamma \equiv$ DENSITY

$H \equiv$ DEPTH OF BUCKET

$\phi \equiv$ ANGLE OF INTERNAL FRICTION

$D \equiv$ BUCKET WIDTH

Actual Digging Forces

This value is much greater than the bearing capacity, but it is the MAX which can be placed on bucket tip.

PASSIVE EARTH PRESSURE ANALYSIS

$K_p = \tan^2(45° + \phi/2)$ $\quad \phi = 37°$

$\Rightarrow K_p = 4.022$

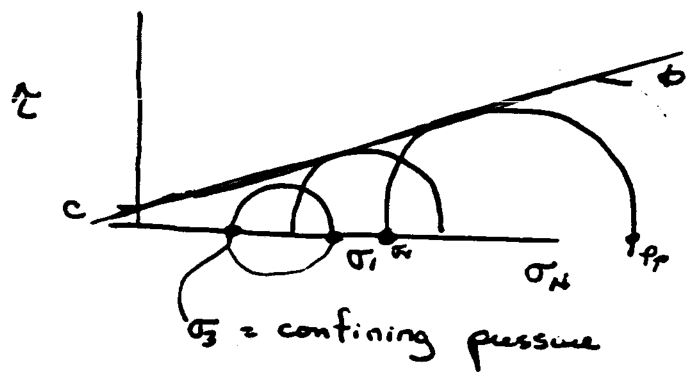
$$P_{passive} = \frac{\gamma H^2}{2} K_p g \left[\frac{\text{FORCE}}{\text{LENGTH}}\right]$$

$$P_p = \frac{(2000\ kg/m^3)(x)^2(1.635\ m/s^2)(4.022)}{2} = 6575\ N_m/m^3$$

$$P_p = (6575\ N/m^3)\ x^2 (.D) = 6575(.7)^2(.2) = 4793\ N_m$$

$\simeq 650\ lb_m$

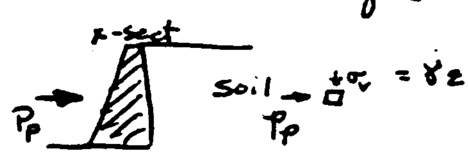
This Analysis shows a better estimation of the actual digging forces on the buckets.



$$\tau = c + \sigma_N \tan\phi$$

cohesion friction angle

Passive Earth Pressure



Lateral Loading Piles Plan →I○→

Pressuremeter - Limit Pressure - P_L & c, ϕ

Impact - VS - Digging Forces

Max Impact Force	-	2800 lb
max Submerging Force	-	6321 lb
max digging Force	-	650 lb

This implies that submerging the bucket will require the most force, but that digging is capable and very realistic. The evaluations above are extreme maximums, and should be used for comparison purposes.

SUMMARY OF LUNAR SOIL PROPERTY VALUES

PROPERTY OR CHARACTERISTIC	PROBABLE VALUE	
	THIS REVIEW	JPL ESTIMATE[a]
SOIL AND SURFACE PROFILE	Fragmental layer of variable thickness. Max. slopes of 33 – 35° (crater sides)	Max. slopes of 34 – 35°
COMPOSITION	Similar to terrestrial iron-rich basalt	—
PARTICLE SIZES	Size Range: boulders to 2μ; bulky particles; varying angularity	2μ – 60μ (fine friction) 50% < 10μ Distribution curves available for 1 mm – 10 m, Surveyor sites
DENSITY UPPER FEW MILLIMETERS BELOW TOP FEW MILLIMETERS	 0.6 – 1.2 gm/cm³ 1.0 – 2.0 gm/cm³ ≅ 2000 kg/m³	 0.7 – 1.2 gm/cm³ 1.5 gm/cm³
COMPRESSIBILITY	Relatively incompressible below top few millimeters (under spacecraft and SMSS loadings)	—
STRENGTH PARAMETERS COHESION ANGLE OF INTERNAL FRICTION	 0.02 – 0.5 psi 33 – 37°	 0.07 – 0.26 psi (0.048 – 0.180 N/cm²) 37 – 39°
BEARING CAPACITY	Increases with depth and breadth of loaded area. A few psi near surface. See text.	Variable with depth Static (average) – 5 psi Upper few mm – 0.15 psi 2 cm depth – 2.7 psi 5 cm depth – 8.2 psi
DYNAMIC PROPERTIES EFFECTIVE SPRING CONSTANT (MODULUS)	7000 psi	—
PERMEABILITY	$1 \times 10^{-8} - 7 \times 10^{-8}$ cm² (Reasonable, but assumptions needed for determination)	$1 \times 10^{-8} - 7 \times 10^{-8}$ cm²

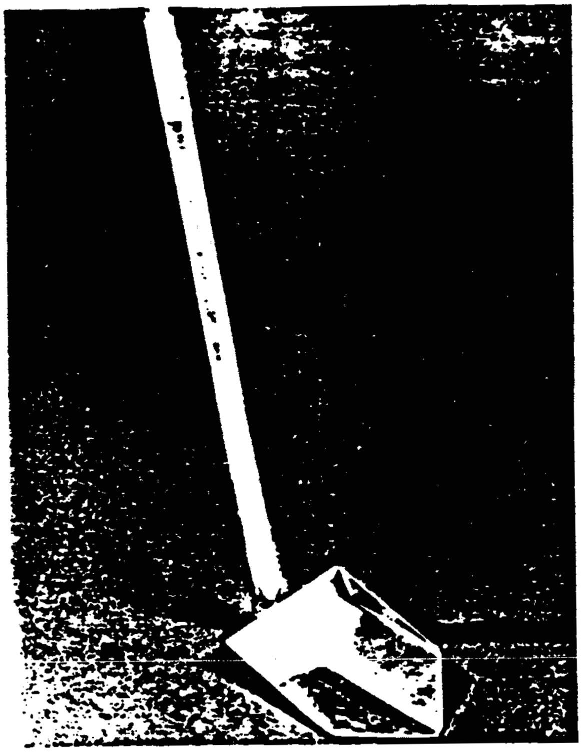


FIGURE 4-15.—Lunar trenching tool (S-70-34925).

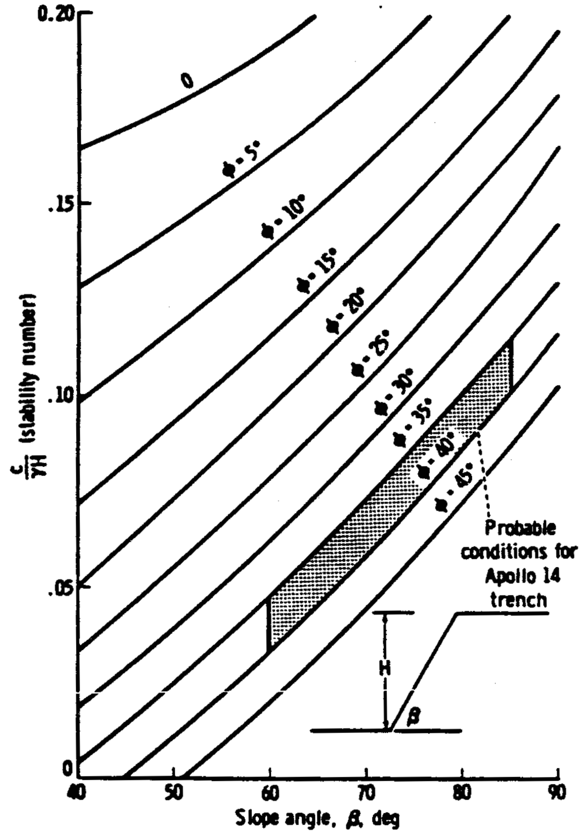
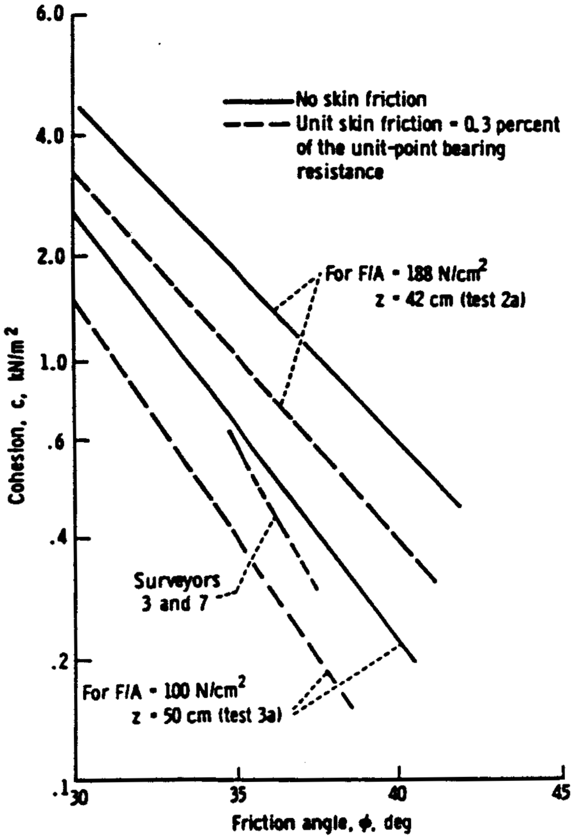


FIGURE 4-23.—Stability numbers for homogeneous slopes.



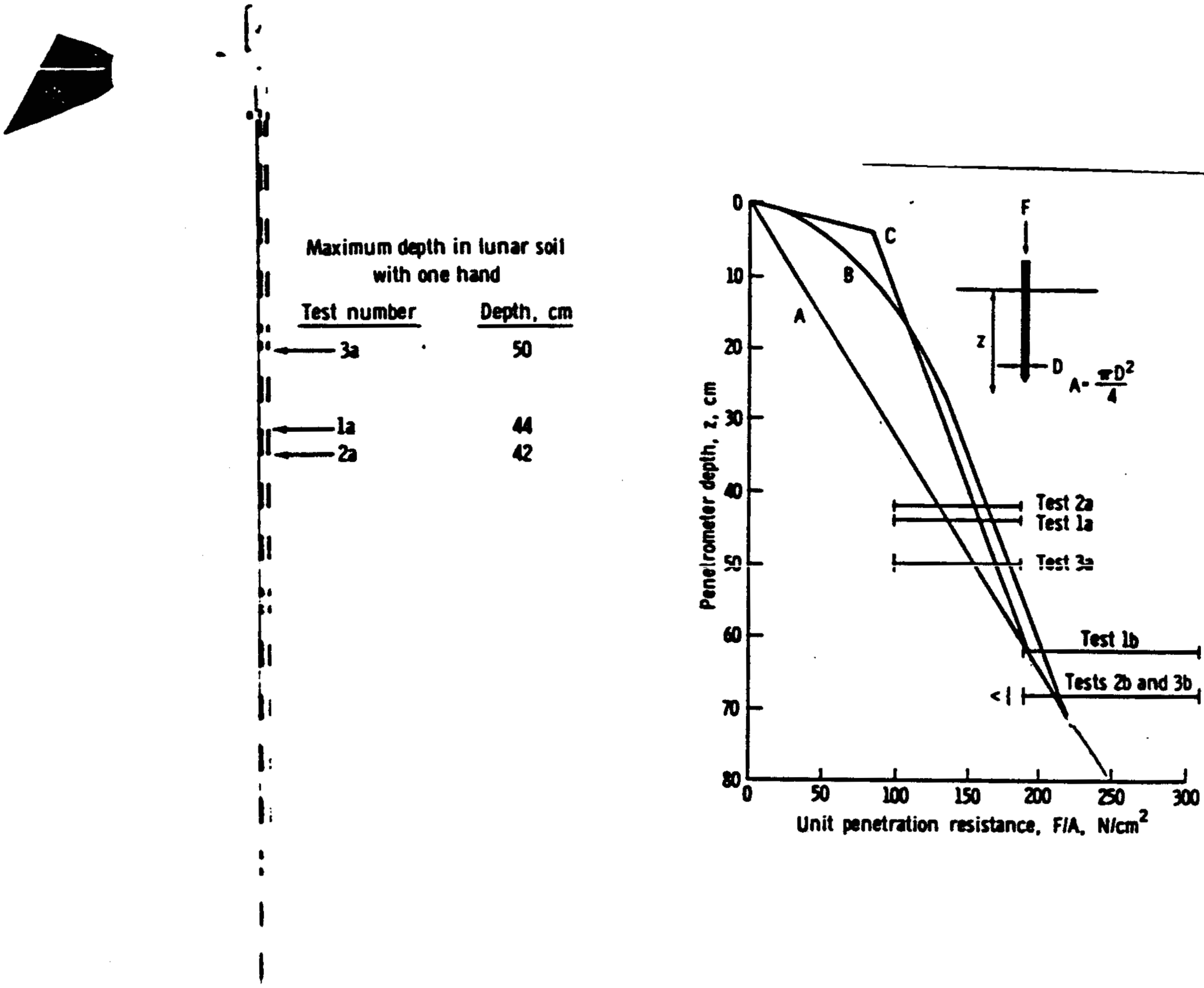


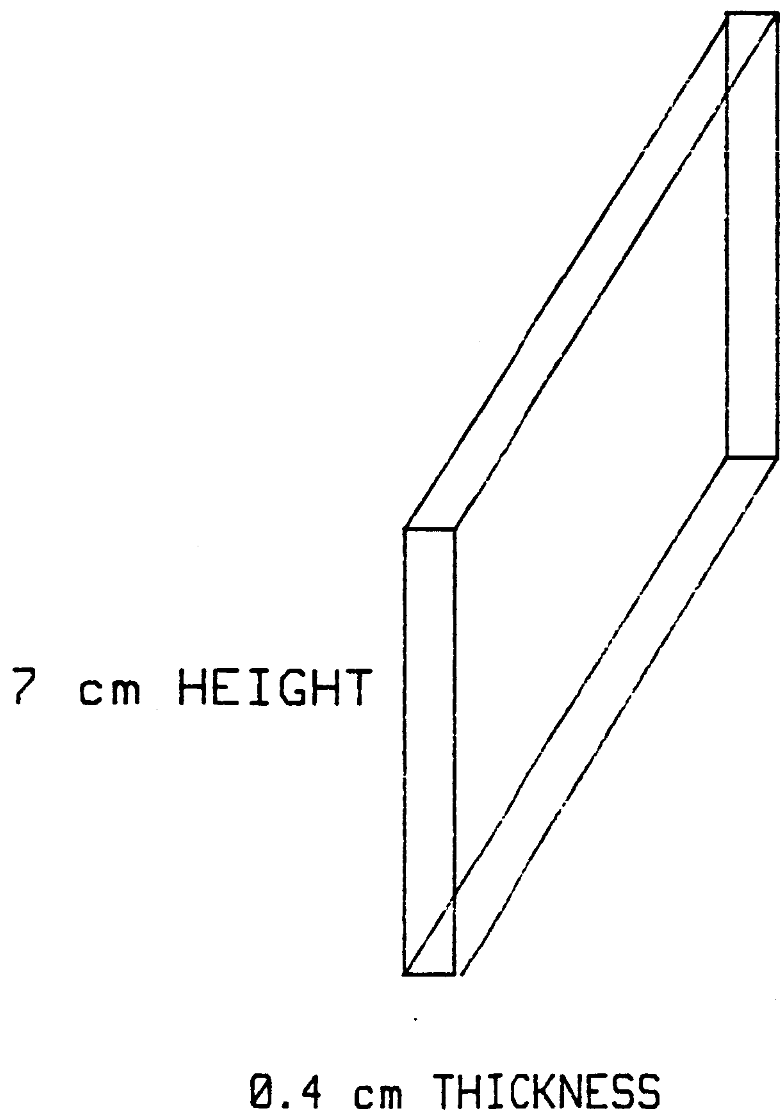
TABLE 4–II. ***ASP Penetrations***

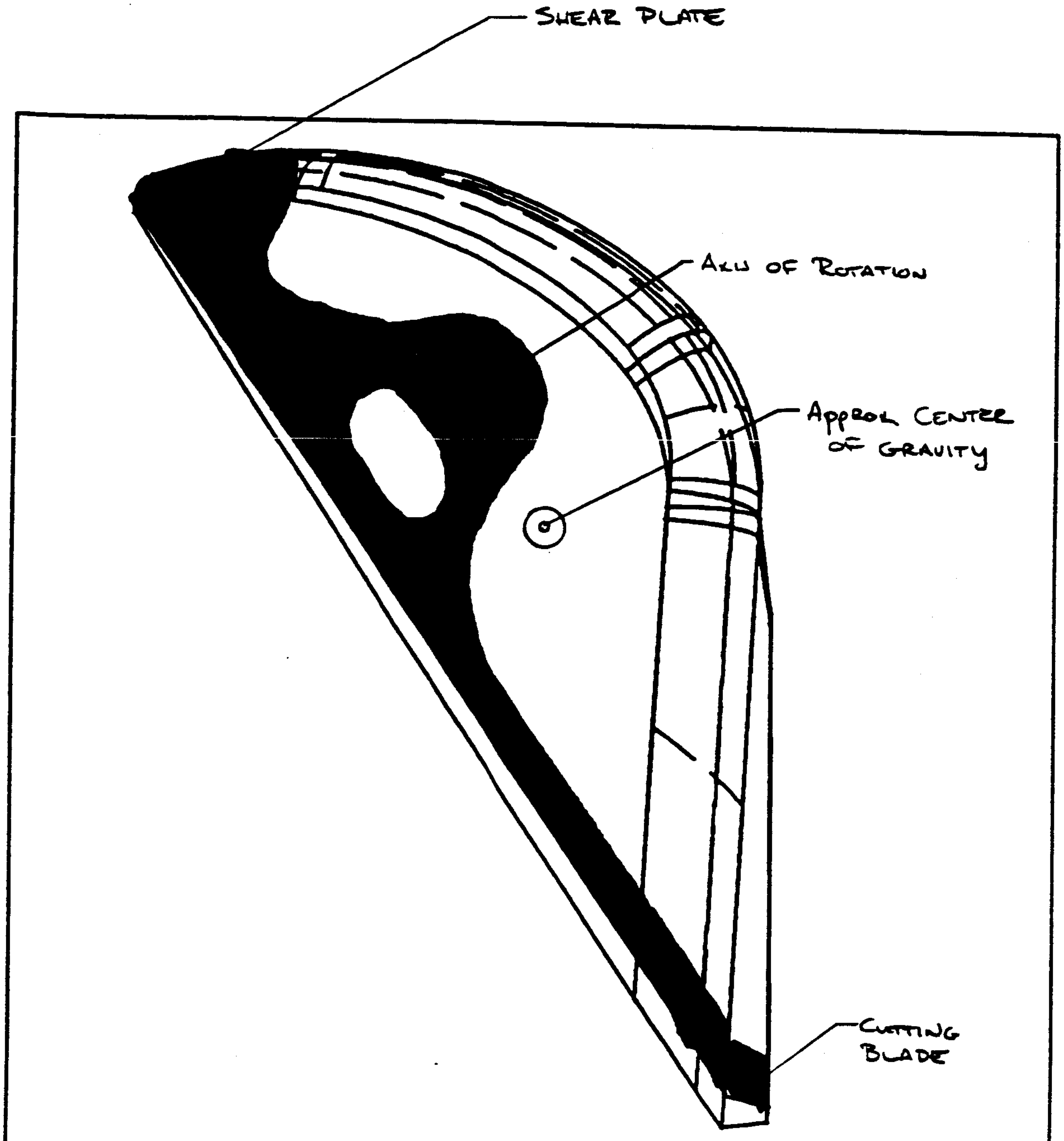
Test	*Depth, z, cm*	*Mode of force application*	*Force, F, N*	*Unit penetration resistance, F/A, N/cm²* [a]
1:				
a..............	44	1-handed	71 to 134	100 to 188
b..............	62	2-handed	134 to 223	188 to 314
2:				
a..............	42	1-handed	71 to 134	100 to 188
b..............	68	2-handed	<134 to 223	<188 to 314
3:				
a..............	50	1-handed	71 to 134	100 to 188
b..............	68	2-handed	<134 to 223	<188 to 314

[a] Cross-section area of the penetrometer, 0.71 cm²; all force is assumed to be carried in point bearing ($F_s = 0$).

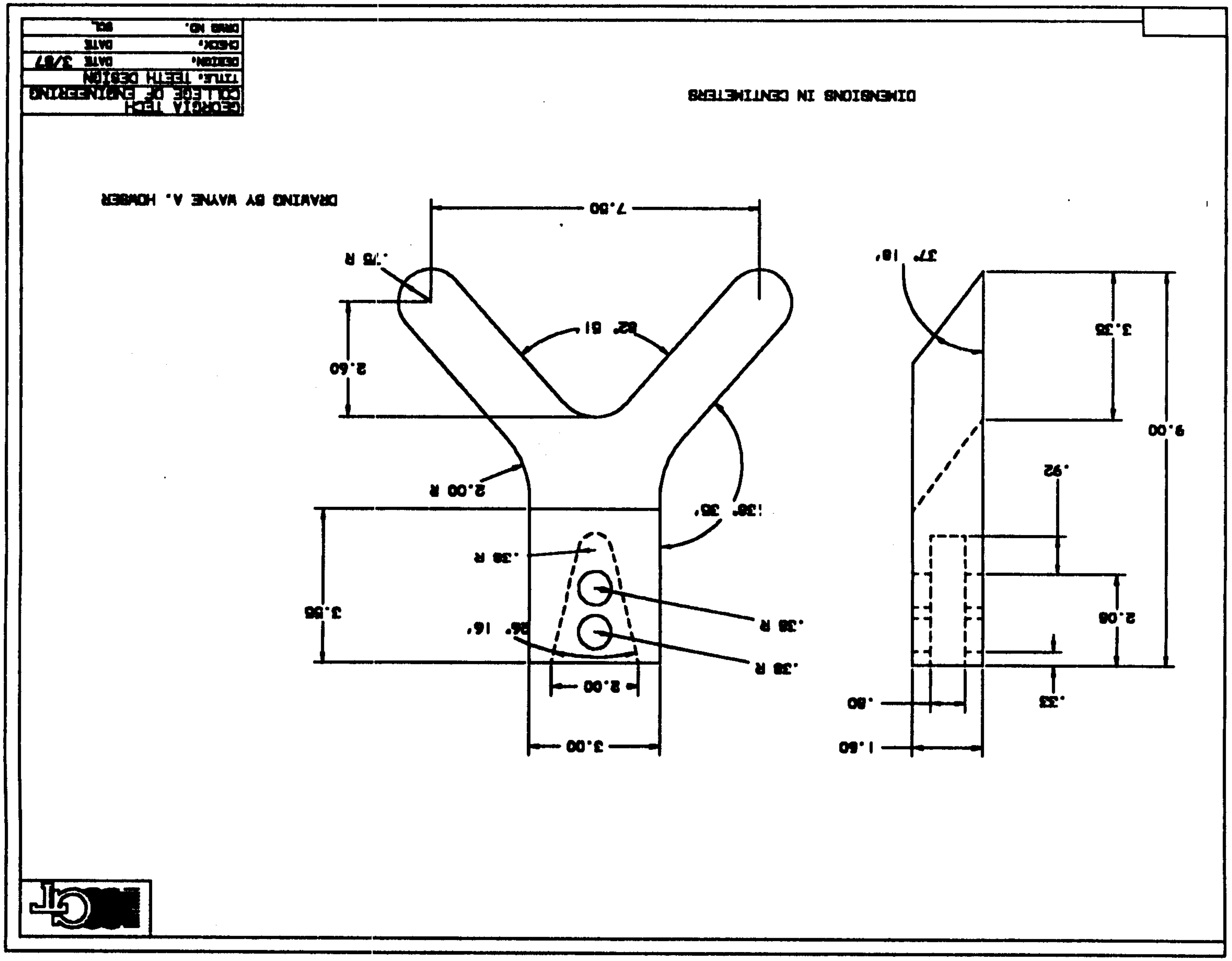
REINFORCEMENT STRUCTURE

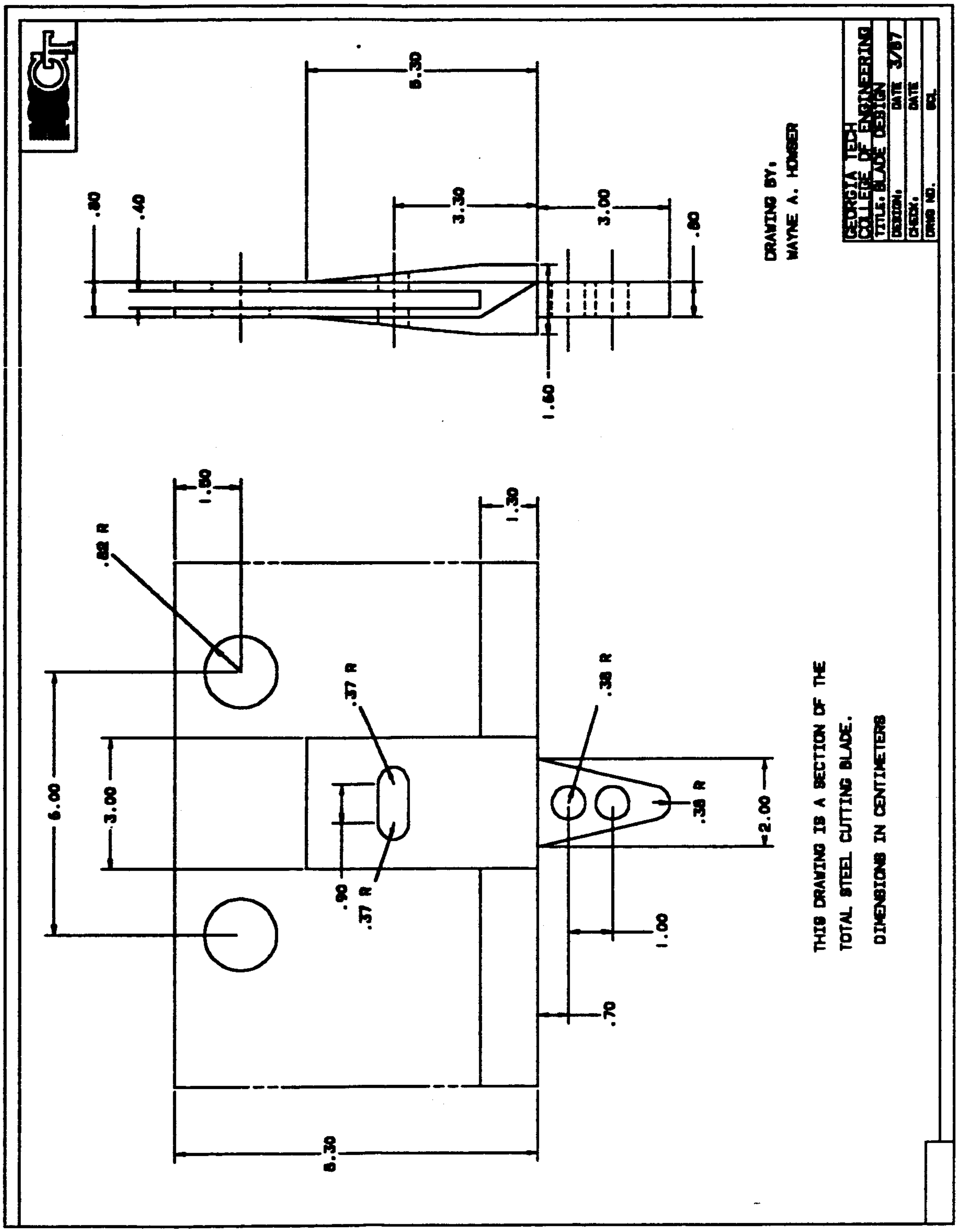
CROSS- SECTION

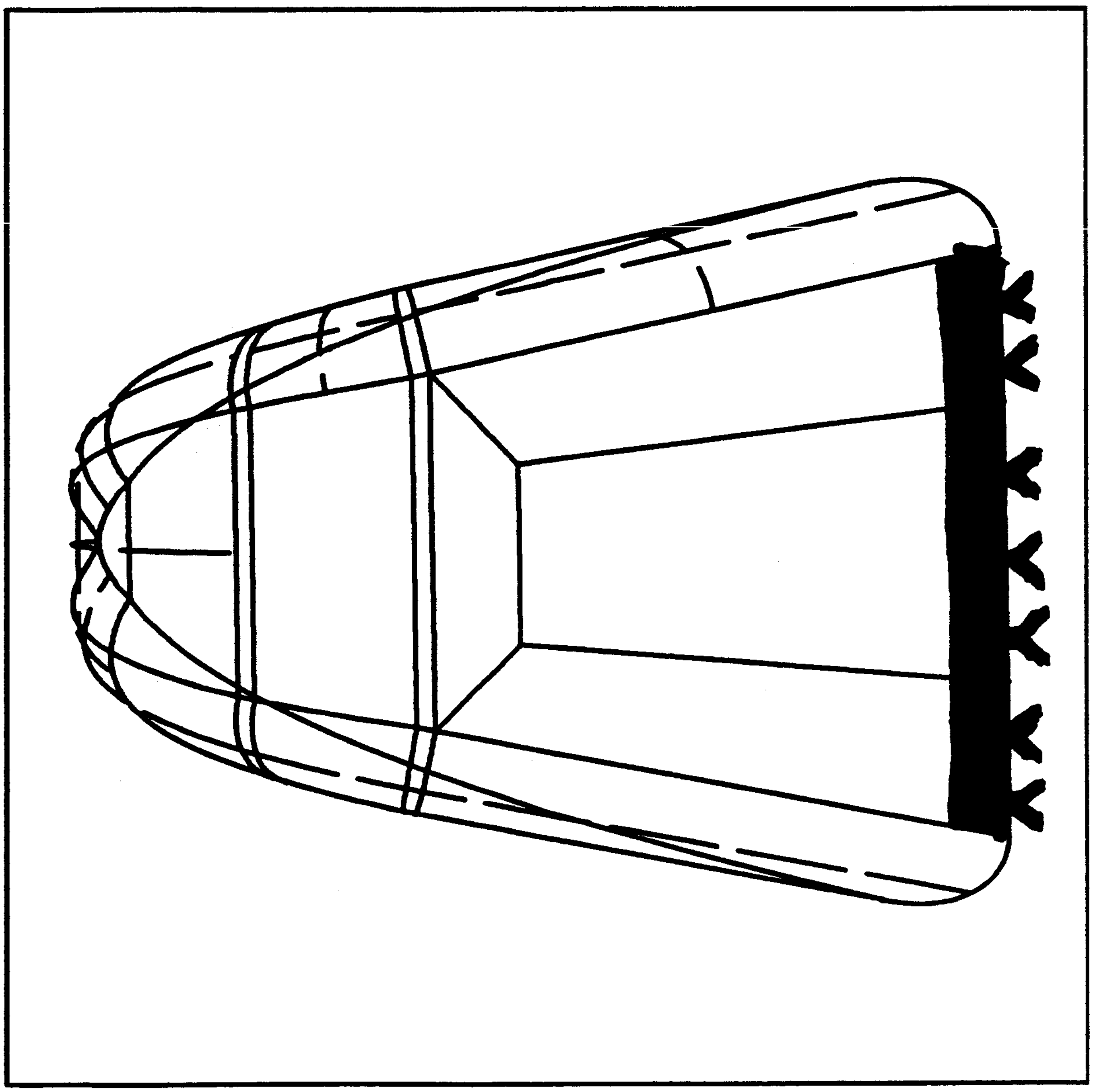




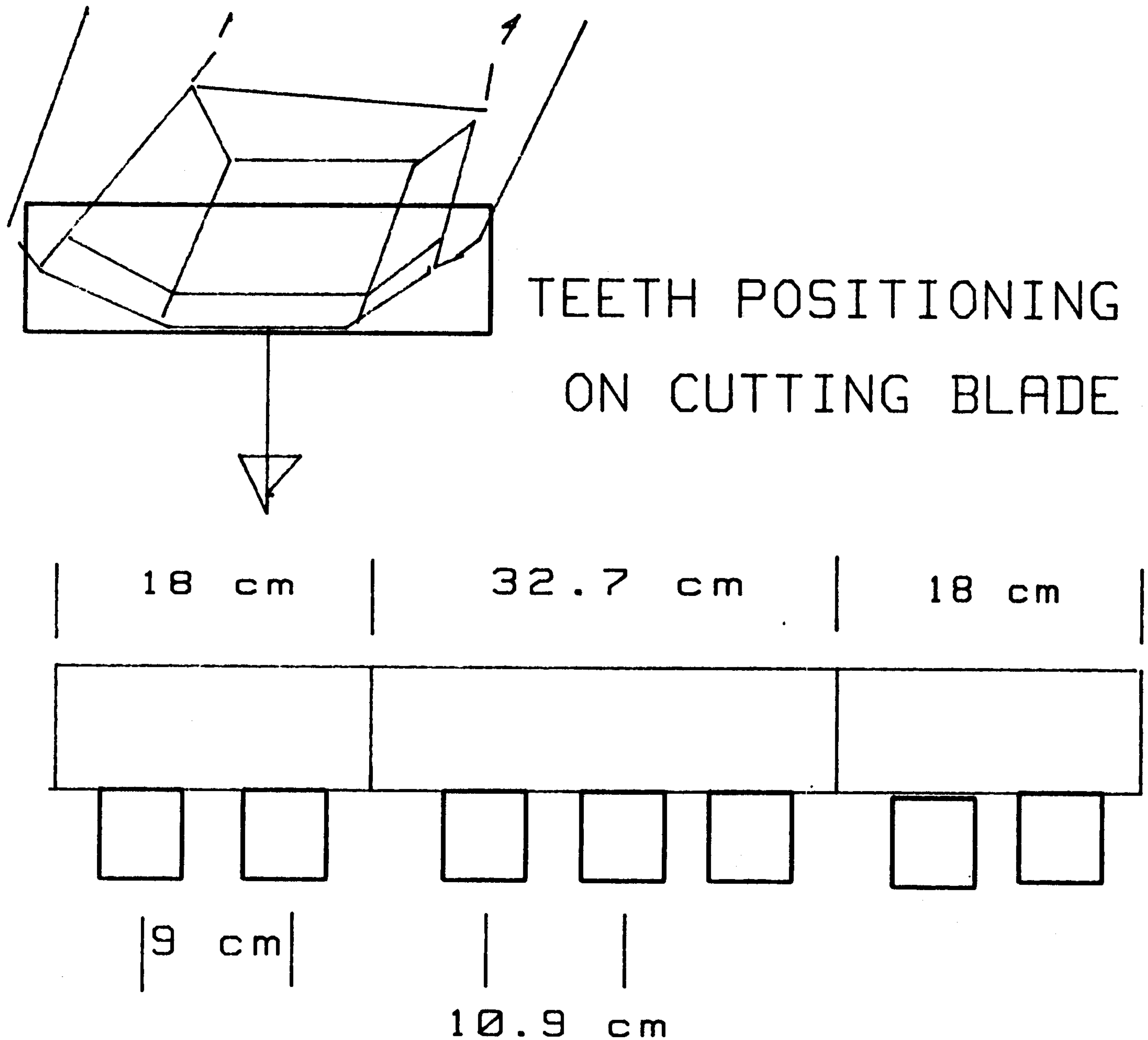
REINFORCEMENT SYSTEM


GEORGIA TECH
COLLEGE OF ENGINEERING
TITLE: TEETH DESIGN
DESIGN: DATE 3/87
CHECK: DATE
DRWG NO. SCL
DIMENSIONS IN CENTIMETERS
DRAWING BY WAYNE A. HOWSER
7.50
.75 R
2.60
82° 51'
2.00 R
136° 35'
.38 R
.38 R
.38 R
3.55
36° 16'
2.00
3.00
37° 18'
3.35
9.00
.92
2.08
.33
.80
1.60

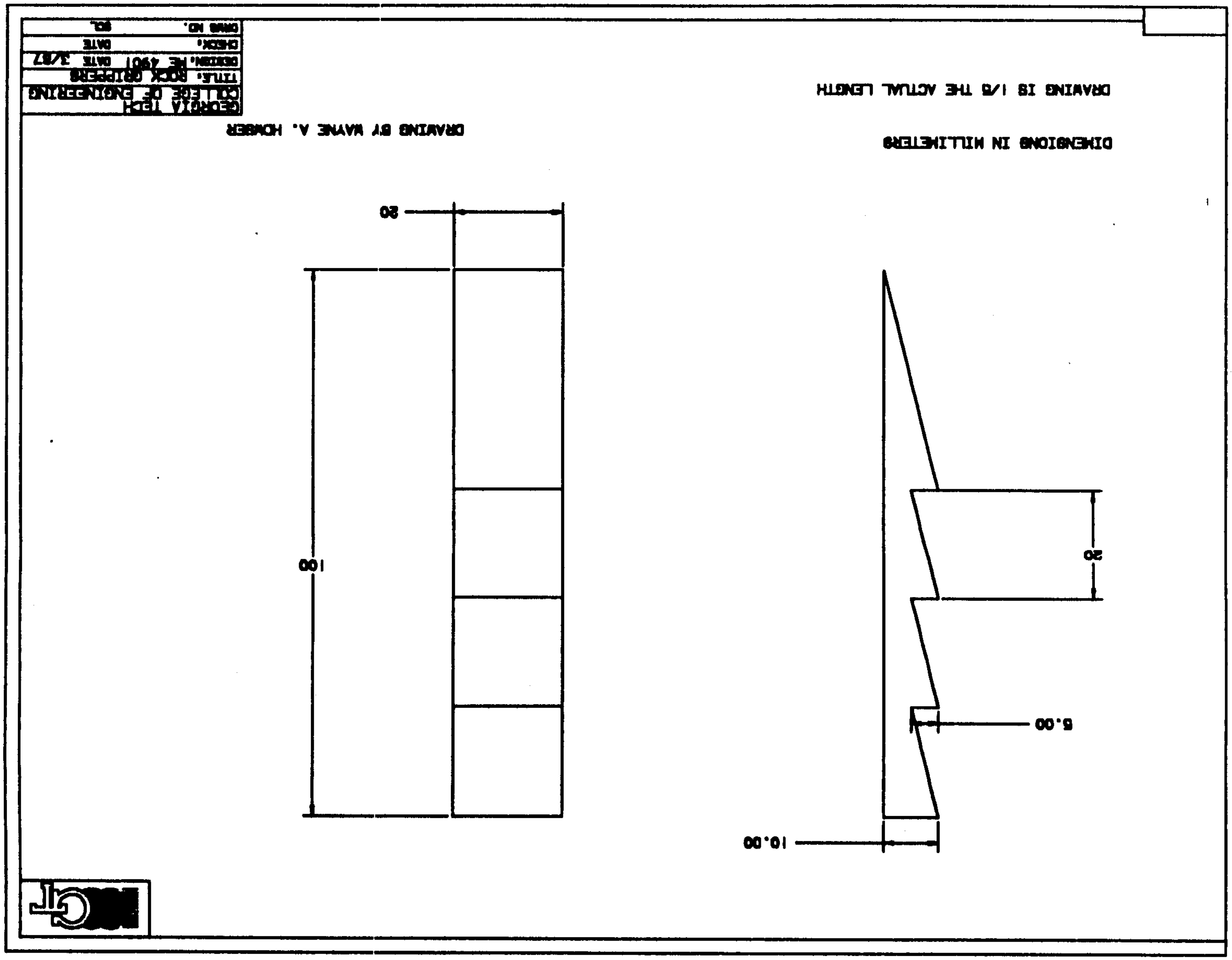

GEORGIA TECH
COLLEGE OF ENGINEERING
TITLE: BLADE DESIGN
DESIGN:
DATE 3/87
CHECK:
DATE
DRWG NO.
SCL
DRAWING BY:
WAYNE A. HOWSER
THIS DRAWING IS A SECTION OF THE
TOTAL STEEL CUTTING BLADE.
DIMENSIONS IN CENTIMETERS
5.30
.80
.40
3.30
3.00
.80
1.60
.82 R
1.50
1.30
6.00
3.00
.37 R
.90
.37 R
.38 R
.38 R
2.00
1.00
.70
5.30



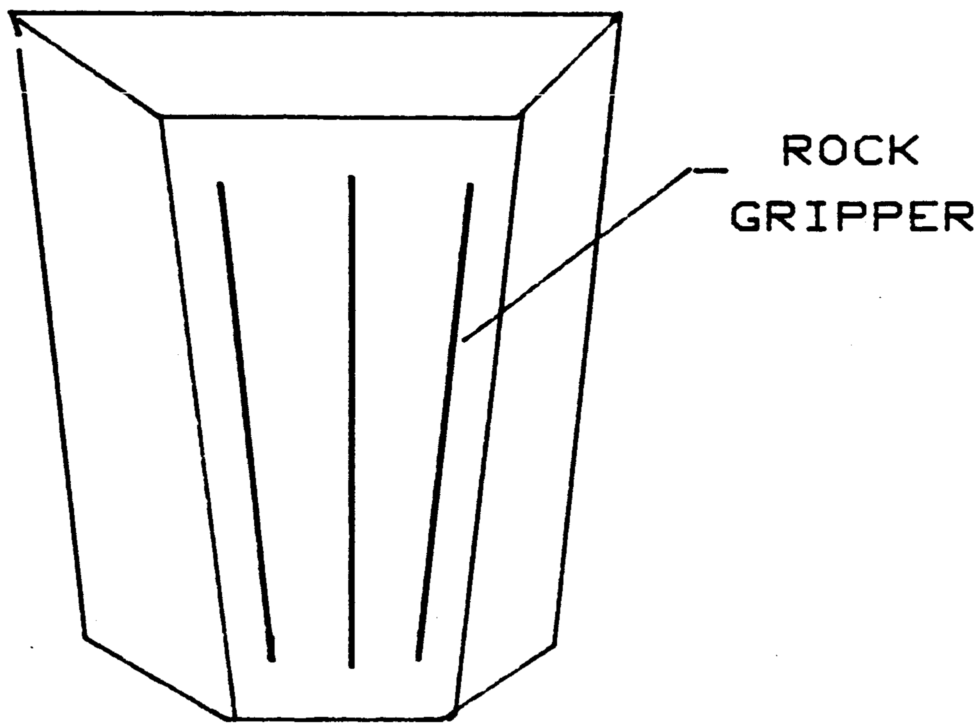
TEETH POSITIONING ON CUTTING BLADE



ROCK GRIPPERS


GEORGIA TECH
COLLEGE OF ENGINEERING
TITLE: ROCK GRIPPERS
DESIGN: ME 4901 DATE 3/87
CHECK: DATE
DRWG NO. SCL.
DRAWING BY WAYNE A. HOMBER
DIMENSIONS IN MILLIMETERS
DRAWING IS 1/5 THE ACTUAL LENGTH
100
20
10.00
5.00
20

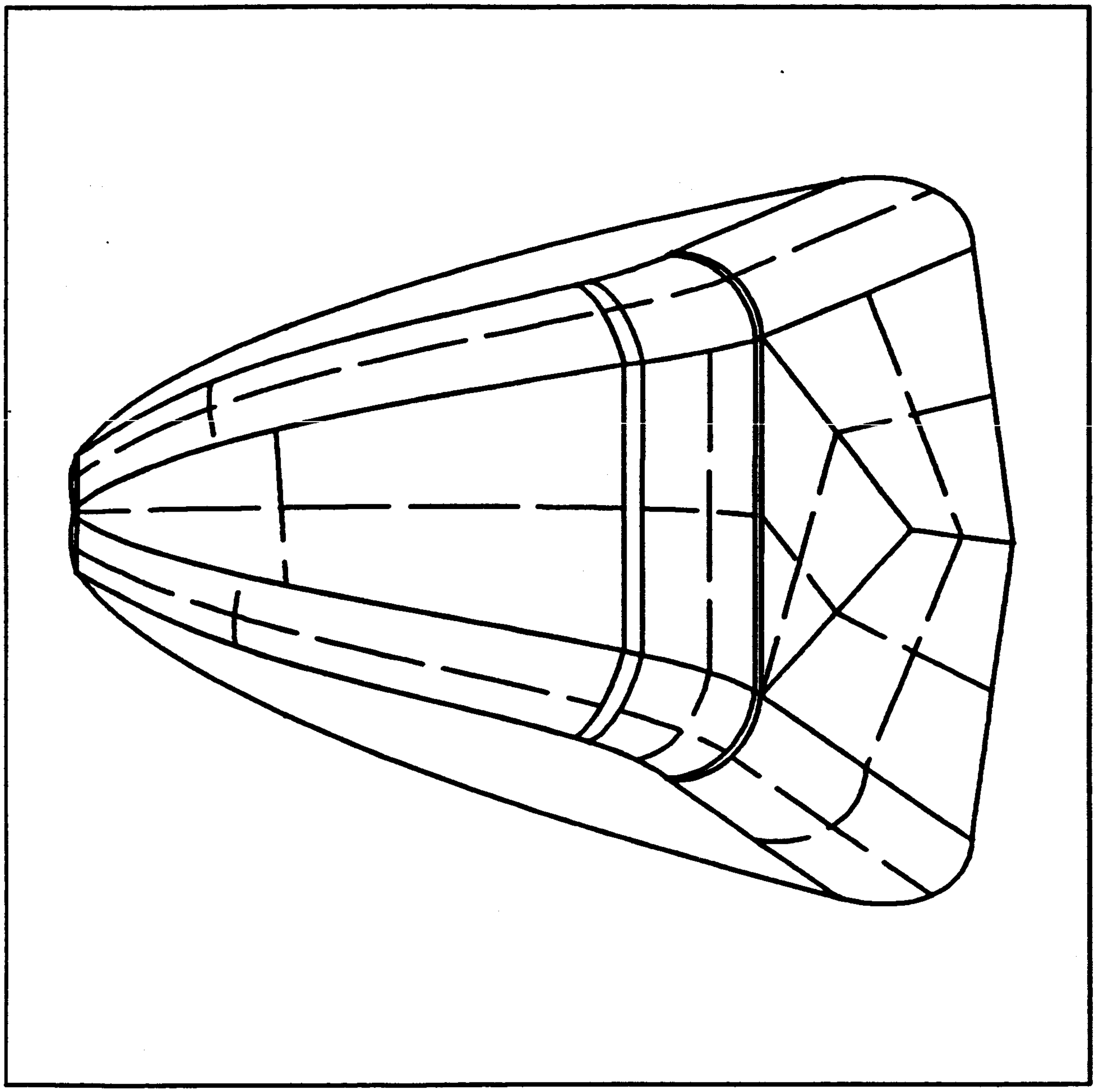
ROCK GRIPPER POSITIONING

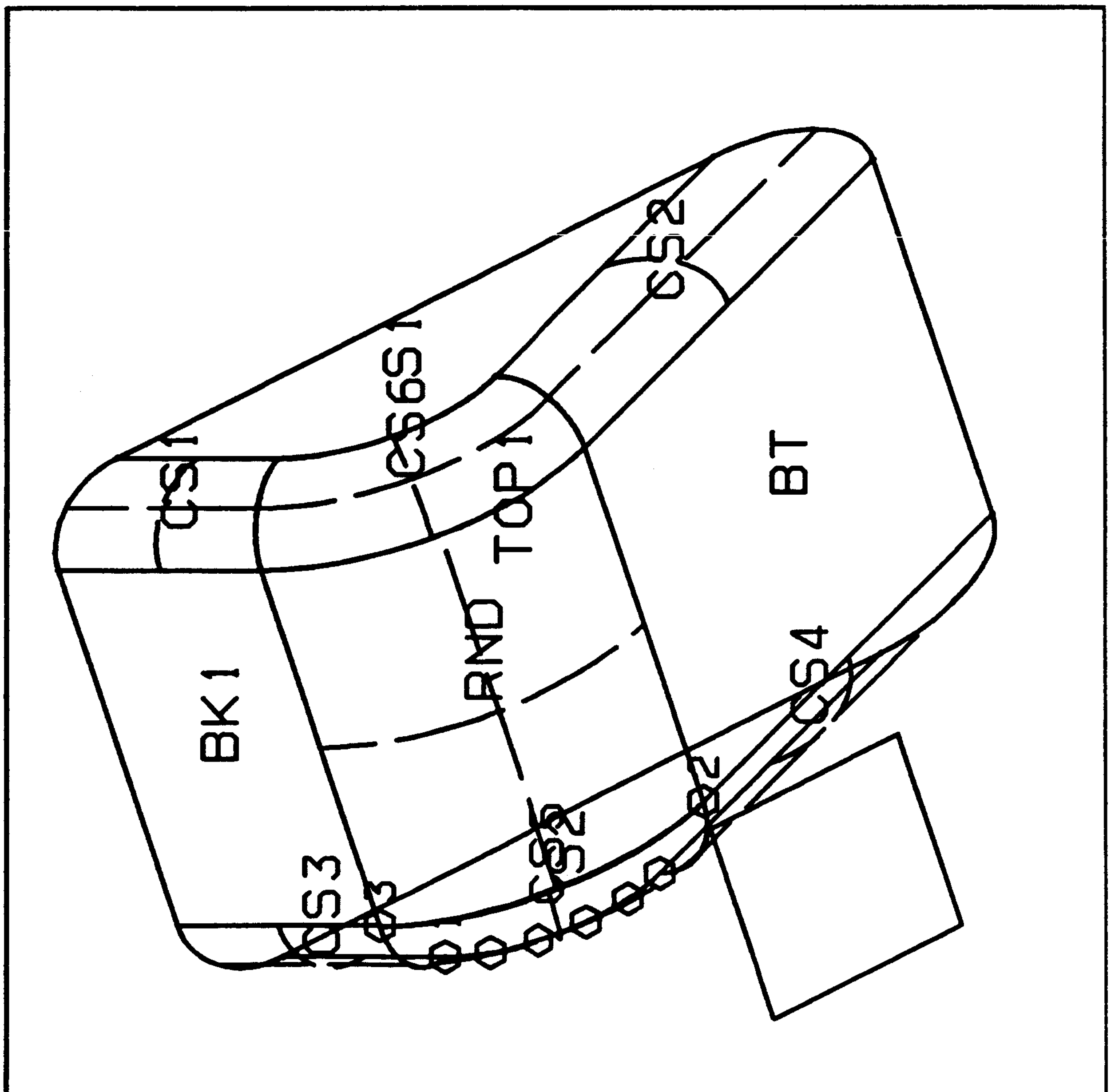


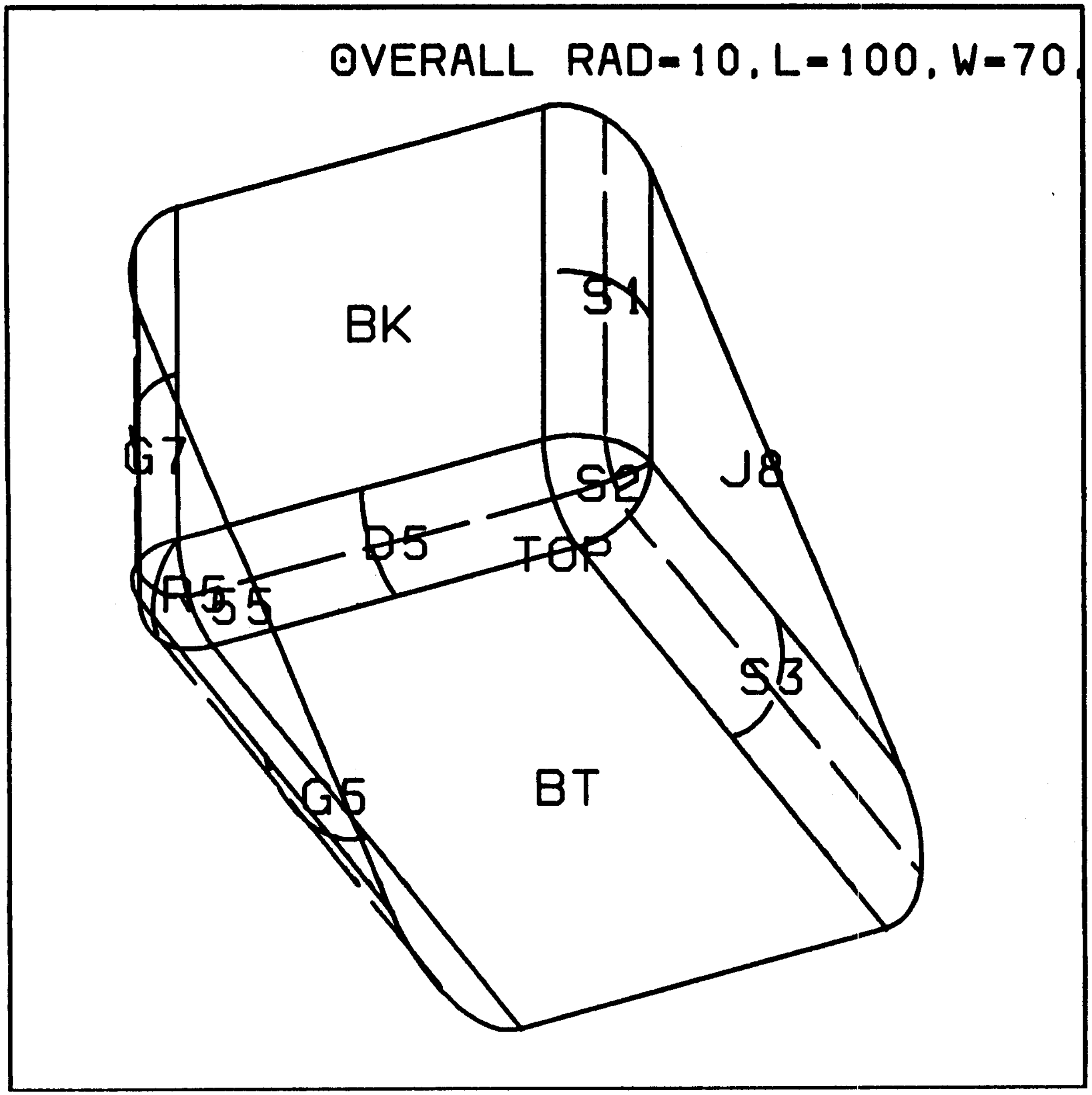
REAR VIEW OF

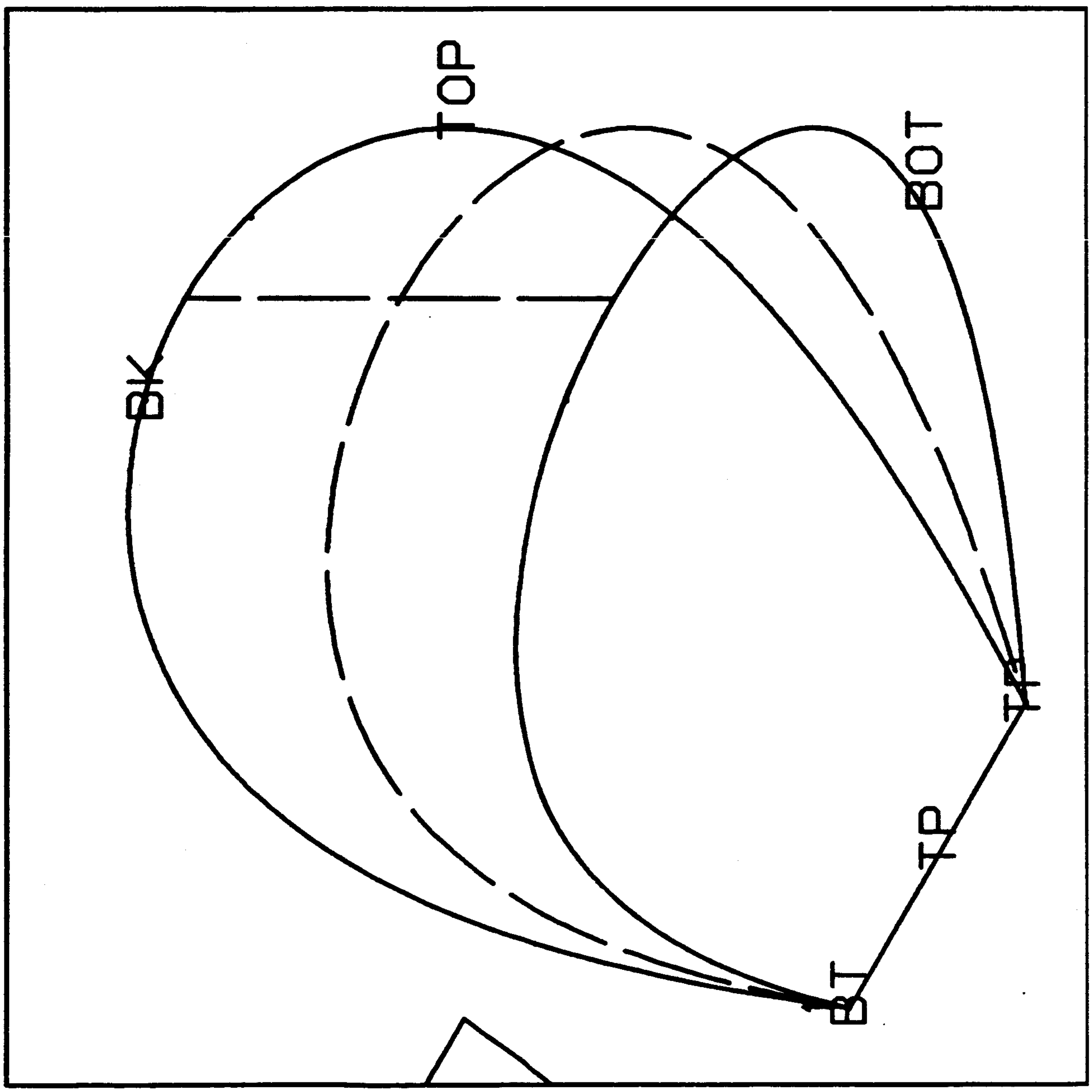
BUCKET UNDERFACE

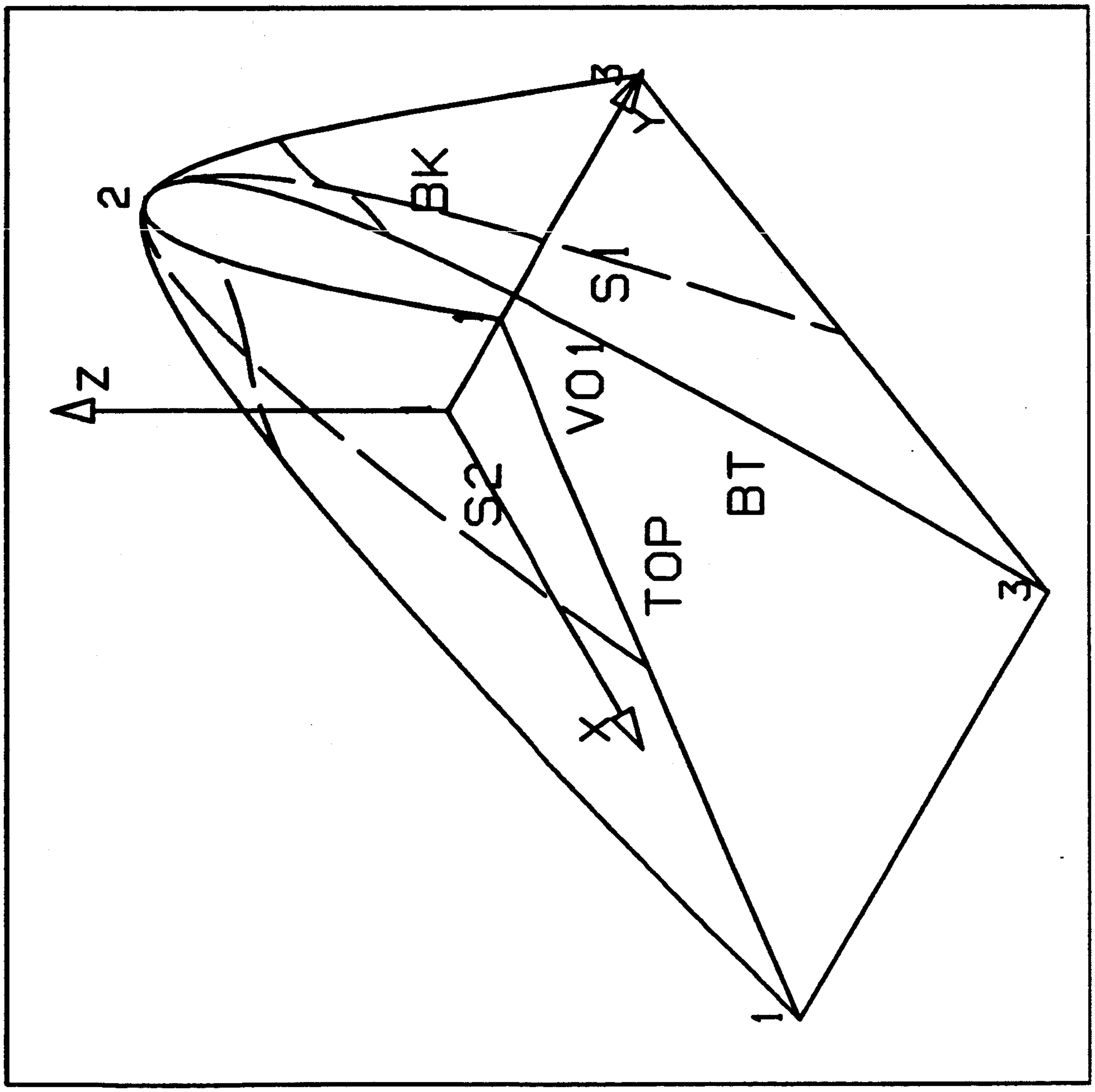
ALTERNATE DESIGNS



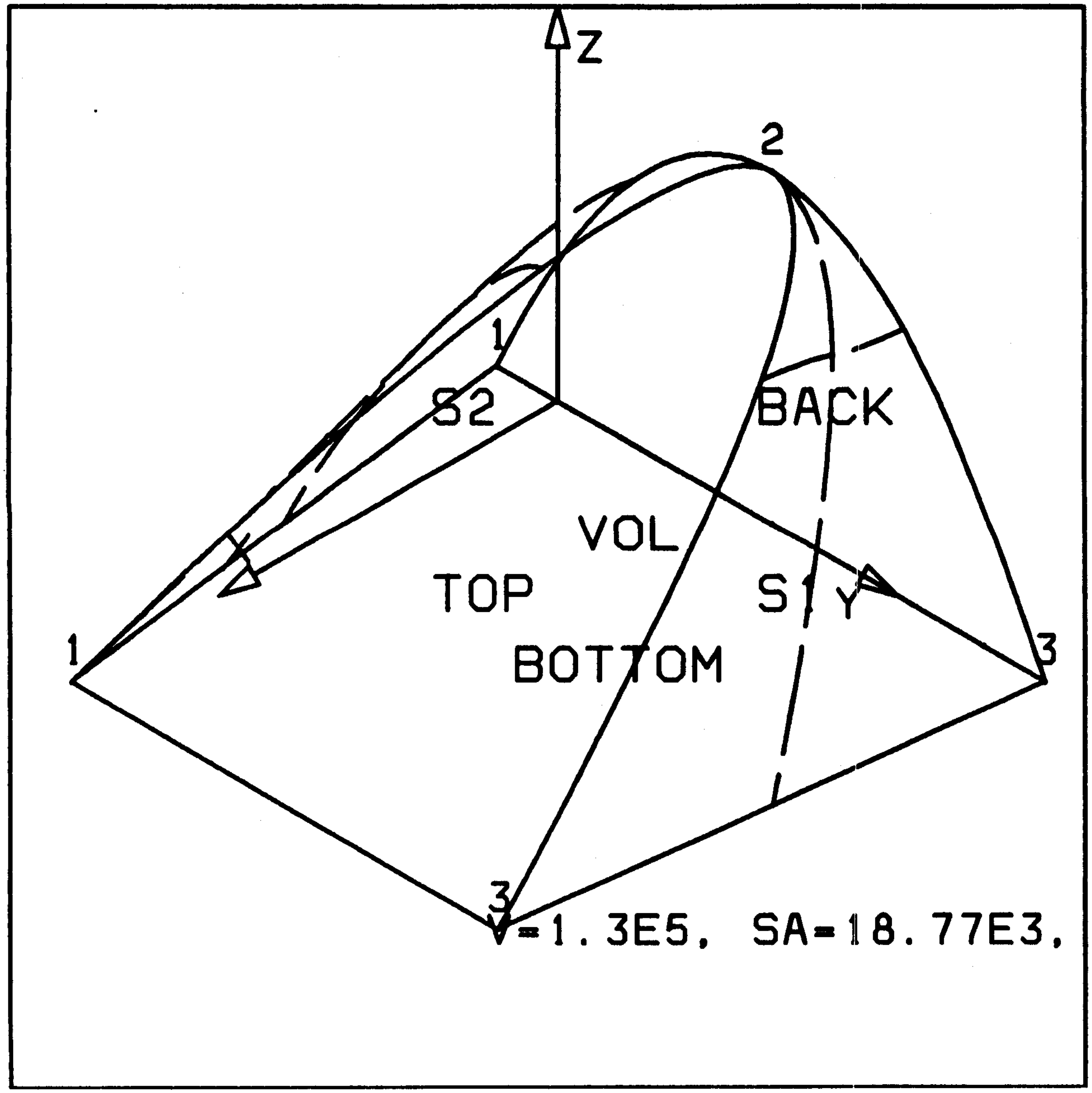

BK1
RND
TOP1
BT
CS1
CS6S1
CS2
CS4

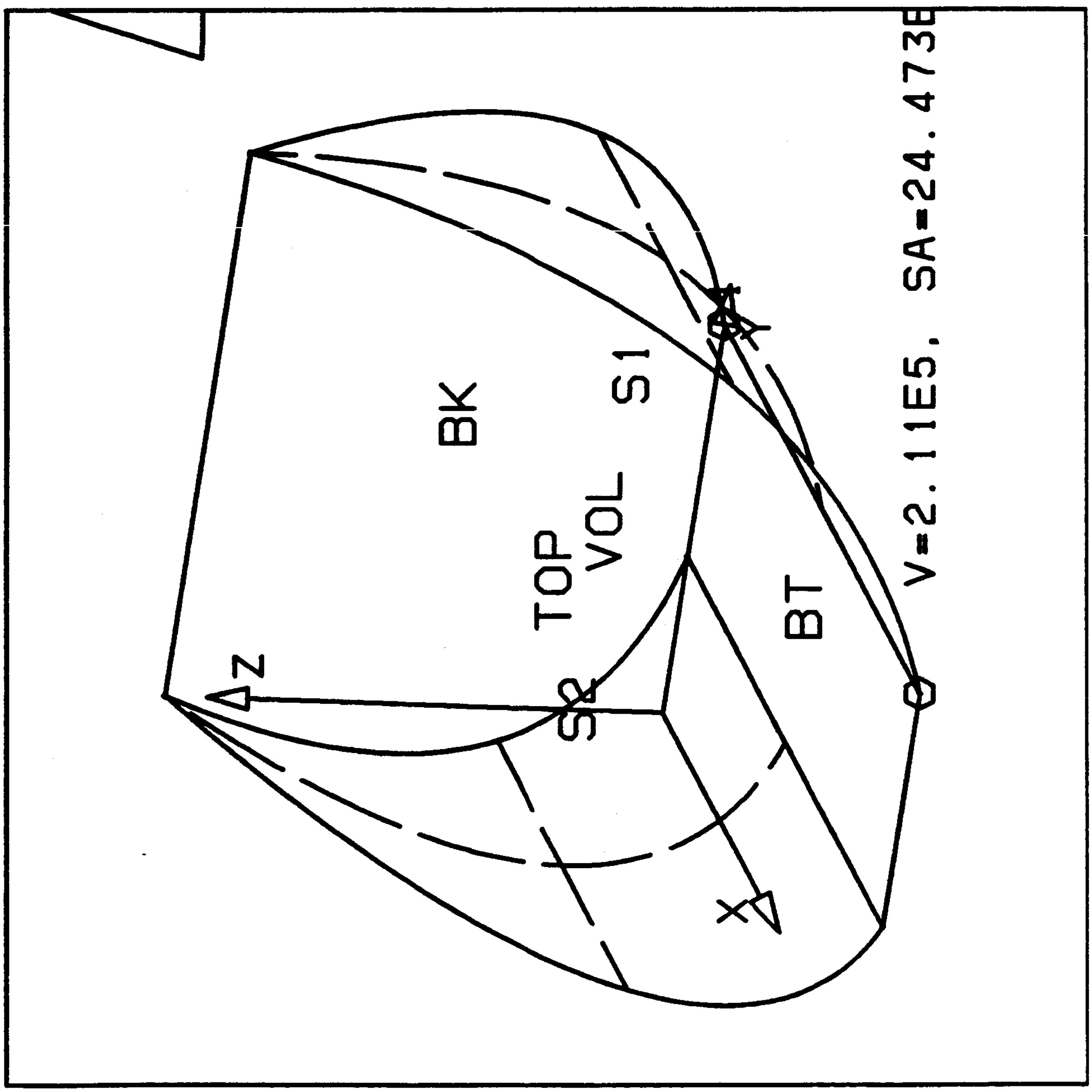

OVERALL RAD=10,L=100,W=70,
BK
S1
G7
J8
S2
D5
TOP
R5
S5
S3
G5
BT


TOP
BOT
BK
TB
TP
BT

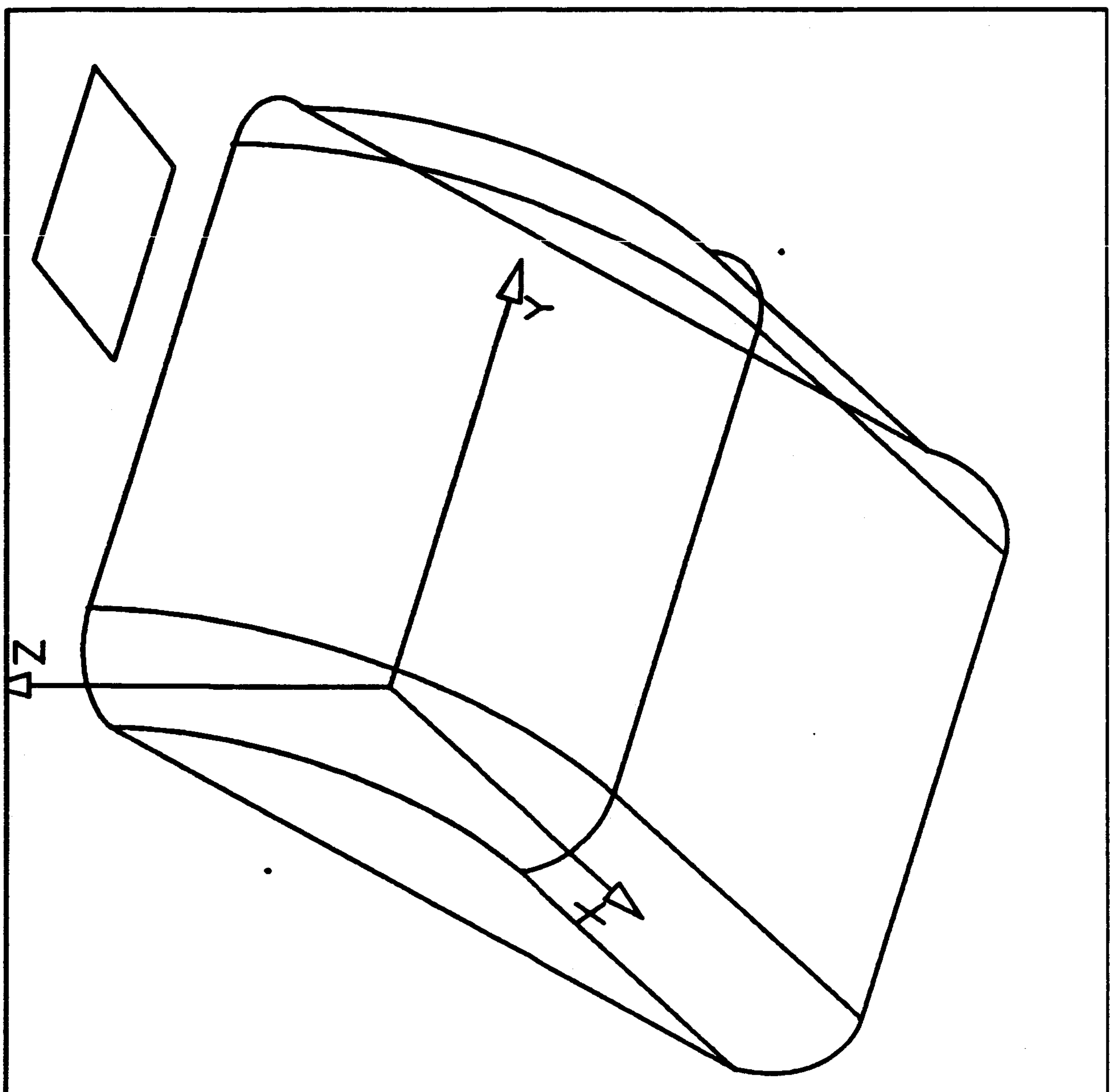


1
2
3
Z
Y
X
BK
S1
S2
VO1
TOP
BT


Z
2
1
S2
BACK
VOL
TOP
S1
Y
1
3
BOTTOM
3
V=1.3E5, SA=18.77E3,



BK
S1
TOP
VOL
S2
Z
Y
X
BT
V=2.11E5. SA=24.473E


Y
Z
X

QUARTERLY CORRESPONDENCE

GEORGIA INSTITUTE OF TECHNOLOGY
George W. Woodruff School of Mechanical Engineering

SPECIAL PROBLEM

INSTRUCTOR : J.W. BRAZELL

LEVEL: UNDERGRADUATE

TITLE: ME 4901, Special Design Project

PROBLEM STATEMENT:

This class will be taken as a design elective to continue working on a design project started in ME 4182, fall quarter, 1986. This project will work towards furthering the development of the LUNAR DIGGER buckets. Several necessary concerns are materials, soil mechanics, bucket geometry, impact forces, and optimization techniques.

FINAL REQUIREMENT: Write-up and presentation

CREDIT 0-3-9

	ACCEPTANCE:	**DATE:**
APPROVAL:	______________________________ Dr. Brighton, Director	_______
STUDENT:	______________________________ Wayne A. Howser	_______
INSTRUCTOR:	______________________________ Mr. J.W. Brazell	_______

FINAL REPORT AND GRADE: ___________________________

INSTRUCTOR:	______________________________	_______
DIRECTOR:	______________________________	_______

Special Project Update
Jan 15, 1987

Wayne Howser

Currently researching in the following areas:

Soils: Found several books in GT library on lunar soils. but I am still looking for one which describes in detail sub-surface soils characteristics. Such as bearing capacity, impact data, etc.

Optimization: Currently reading about several types of optimization processes.

Constraints: Have been looking into finding professionals who have experience in designing excavating buckets to assist with design criteria and tricks of the trade.

Materials: Reading up on materials selection procedures. but the actual selection will probably be about 2-4 week down the road. I feel that more specifics are required on soils before materials can be justly researched.

BUCKET REDESIGN UPDATE #2
1/29/87

WAYNE HOWSER

PROBLEM STATEMENT: REDESIGN THE LUNAR DIGGER'S BUCKETS TO OPTIMIZE THEIR EFFECTIVENESS IN THE LUNAR SOIL. THIS REDESIGN WILL BE CONCERNED WITH MATERIAL SELECTION, BUCKET GEOMETRY, SOIL MECHANICS, AND OPTIMIZATION TECHNIQUES.

MATERIALS: TALKED WITH DR. MEYERS AND I'M STARTING TO NARROW THE FIELD ON MY SELECTION. SOME POSSIBLE MATERIALS:

Ti-3 Al-13 V-11 Cr(a,b) - HIGH STRENGTH TO WEIGHT RATIO AND GOOD ELEVATED TEMP CHARACTERISTICS, BUT CAN'T FIND ANY LOW END TEMP DATA.

MAGNESIUM: DIFFICULT TO FORM, GREAT STRENGTH TO WEIGHT, MODERATE ELONGATION IF HEAT TREATED, BUT TEND TO BE BRITTLE.

COMPOSITES: RESEARCHING THIS. ONE PROBABLE IS ALUMINUM WITH SILICON CARBIDE IN THE WHISKER OR PARTICULAR FORM. THIS MIGHT BE A GOOD SOLUTION.

RADIATION EFFECT: HAVE READ IN SEVERAL TEXTS AND DR. MEYERS CONFIRMED THAT RADIATION HAS LITTLE EFFECT ON MATERIAL PROPERTIES.
APPROX. RAD ENERGY ~ [illegible] MeV

SOILS: CAN ACQUIRE SIEVES, NEED YOU (MR. BRAZELL) TO LOCATE SOME BASALT AND WE ARE IN BUSINESS FOR SOME SIMULATED SOIL.

GEOMETRY: LEARNING VERSACAD, WILL TRY TO GET SOME MODELS UP FOR NEXT WEEK.

DIGGER BUCKET UPDATE
2 - 5 - 87
WAYNE A. HOWSER

MATERIALS: HAVE ACCUMULATED SOME DATA ON POSSIBLE MAGNESIUMS.
NEED TO RESEARCH FURTHER ON TITANIUM, AND ALUMINUM.

CURRENT FEELINGS: ALUMINUM - TO DUCTILE AT HIGH AND LOW TEMPS.

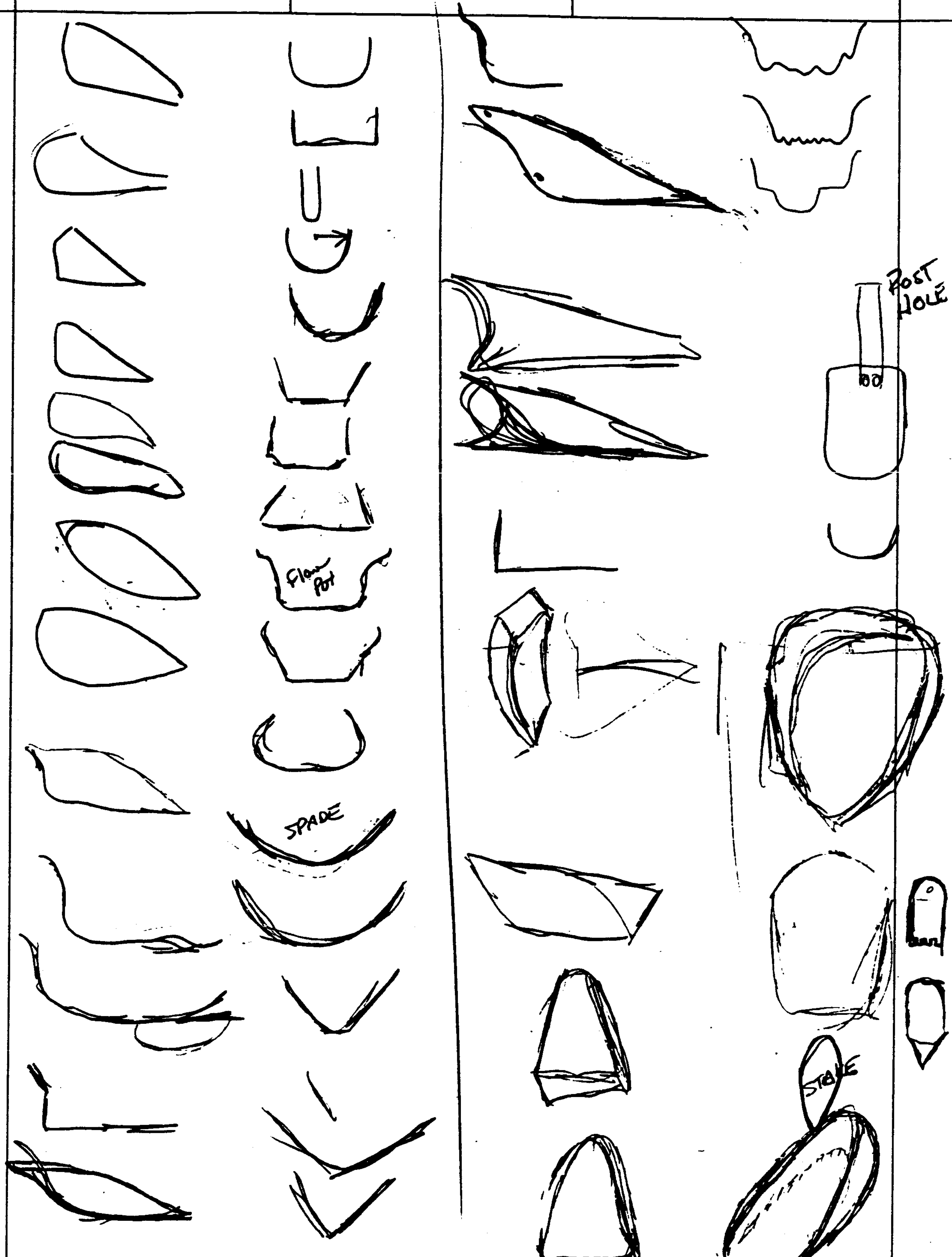
TITANIUM - BEST STRENGTH TO WEIGHT & MOD OF ELAS. TO WEIGHT. LOOKING FOR LOW TEMP DATA.

STILL HAVE NOT LOCATED DR. COLTON TO SEE ABOUT MATERIALS SEARCH.

SOILS: HAVE FOUND SOME GOOD BEARING CAPACITY DATA WHICH WILL AID IN INITIAL BUCKET FORCES. ALSO, FOR ANGLE OF CUT - NEED SOME INSIGHT INTO SOME FIGURES I HAVE FOUND.
ANY LUCK WITH THE BASALT ????

GEOMETRY: DREW UP A BRAIN STORMING SESSION, AND HAVE LEARNED THAT IBM CADAM HAS AN AREA CALCULATING FUNCTION.
I HAVE ALSO FOUND SOME HISTORICAL AGRICULTURAL TOOLS.

NATIONAL
42-381 50 SHEETS 5 SQUARE
42-382 100 SHEETS 5 SQUARE
42-389 200 SHEETS 5 SQUARE



DIGGER BUCKET UPDATE
WAYNE A. HOWSER
2 -12- 87

MATERIALS: Still researching - found some good info on aluminums, but I can not find anything on titanium.

Have spoken with Dr. Colton and his is copying his MATERIALS SELECTION SEARCH for me to use.

I plan to narrow my choices by next week.

GEOMETRY: Learning CATIA. After finally sorting through the possible computer systems, I really think that CATIA is the most appropriate. The only set back is the tutorial is for an old version and the learning process is slow.

I plan to have AT LEAST 5 -6 geometries in by next week and a hard copy of them.

OTHER CONCERNS:

* Attaching buckets to end effector
* End effector refinement if time allows
* soil simulant
* Best method to manufacture buckets
* Functionability of buckets

SOIL MECHANICS: Force evaluation will be done once I get a better idea of what the geometry will look like. I was the one who did the analysis last quarter so this will not pose a problem.

* I need to find a rough estimate of the SKITTER weight for this analysis.

Tentative Schedule

2/19	Have design in catia material selections picked - 3 or 4
2/26	Pick several "good buckets" Start force analysis with F.E.A.
3/5	Finish FEA and tie up loose ends
3/11	Report due and presentation

Bucket Design Update
2-26-87
Wayne A. Howser

MATERIALS: for the bucket Aluminum 2014-T4
for the wear sleeve steel(?)

WEAR PROTECTER I have devised a design for a sleeve type device which can easily be placed on the bucket and will absorb the majority of the wear.

GEOMETRIES, VOLUMES: I have put numerous hours into declaring surfaces which in turn define volumes, but my lack of time and CATIA expertise is becoming too evident. I do have several "primitive" volumes declared, But as soon as the design becomes slightly complicated trouble arises. But by using CATIA, I have gained some insight into these geometries and feel my final design is in view.

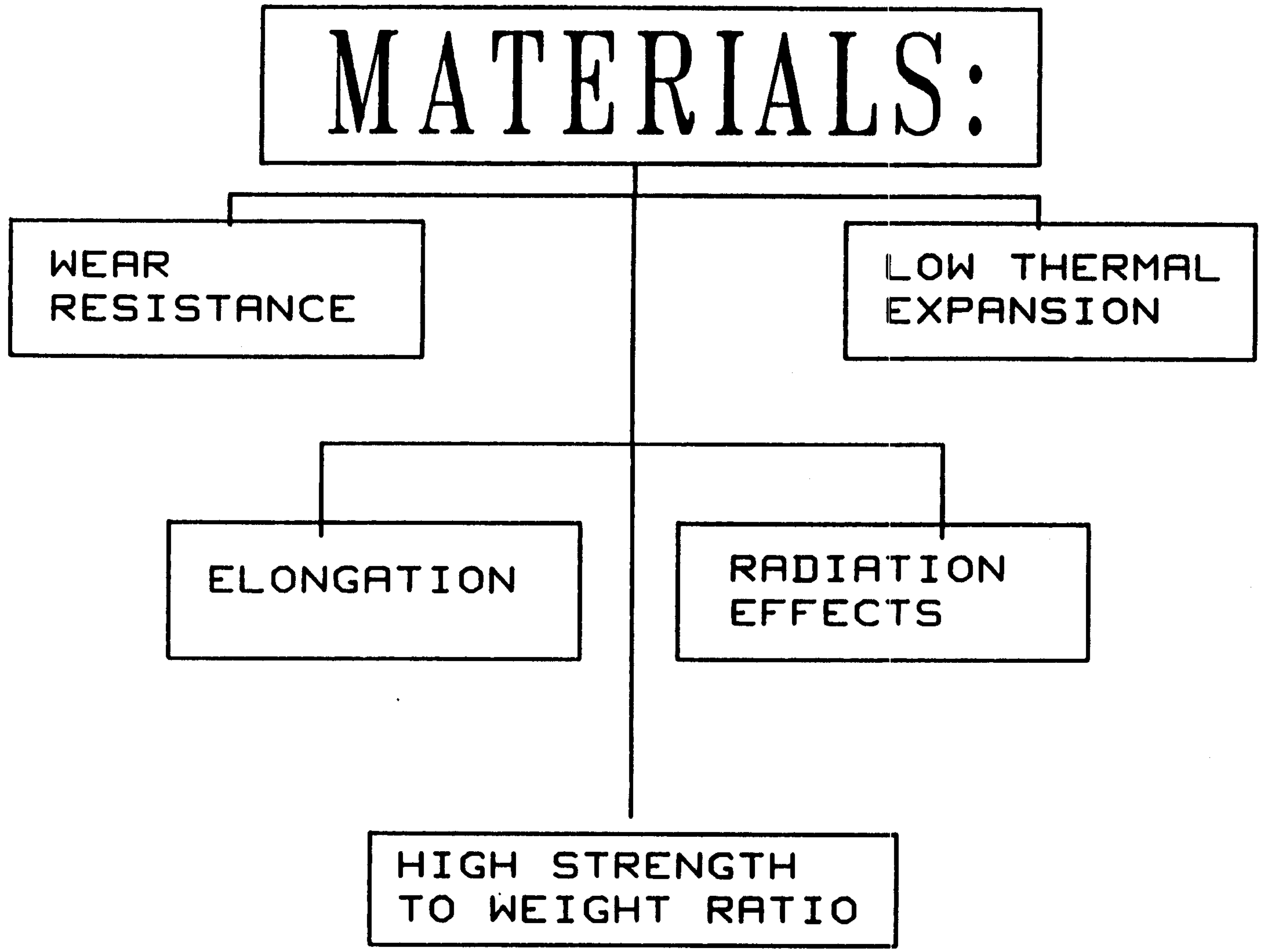
ATTACHMENT TO END EFFECTOR : Coordinating activities with the digger group and feel that the attachment of this design to theirs is possible.

TEETH -vs- BLADE: Looking into designing a cutting edge for the wear protector which will have good surface finishing and breaking features.

PROBLEM STATEMENT:

TO REDESIGN THE LUNAR DIGGER'S BUCKETS TO OPTIMIZE THEIR EFFECTIVENESS IN THE LUNAR SOIL.

CONCERNING: MATERIAL SELECTION, BUCKET GOEMETRY, SOIL MECHANICS, AND OPTIMIZATION TECHNIQUES.


MATERIALS:
WEAR RESISTANCE
LOW THERMAL EXPANSION
ELONGATION
RADIATION EFFECTS
HIGH STRENGTH TO WEIGHT RATIO

PARAMETERS	VALUES
CAPACITY	0.5 FT (+)
WEIGHT	260 E. lbs (−)
BEARING CAPACITY	1 − 1.2 Kgf/cm
OPTIMUM ANGLE OF CUT	Θ
LENGTH, WIDTH, AND DEPTH	X, Y, Z

DESIGN MATRIX

$$
\begin{array}{l}
\text{MATERIALS} \\
\\
\text{GEOMETRY} \\
\\
\text{SOILS} \\
\\
\text{OPTIMIZATION}
\end{array}
\left[
\begin{array}{ccccccccc}
0 & 0 & 0 & 1 & 0 & 1 & 0 & 1 & 0 \\
\\
1 & 0 & 0 & 0 & 1 & 0 & 0 & 1 & 1 \\
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1 & 1 & 1 & 0 & 0 & 1 & 1 & 0 & 1 \\
\\
0 & 1 & 1 & 1 & 0 & 1 & \cancel{0}\,1 & 0 & 1
\end{array}
\right]
\begin{array}{l}
\text{ANGLE OF CUT} \\
\text{L, W, D} \\
\text{CAPACITY} \\
\text{WEIGHT} \\
\text{BEARING CAPACITY} \\
\text{STABILITY (STRUCTURAL)} \\
\text{INTERNAL RADIUS} \\
\text{ENVIRONMENT} \\
\text{IMPACT FORCES}
\end{array}
$$